MANUAL FOR THE DESIGN OF FERROUS AND NON-FERROUS PRESSURE VESSELS AND TANKS

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By

KARL SIEMON

Third Edition

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Preface

The design of pressure vessels has undergone a great change in the past two decades. The rapid increase in the pressures and temperatures employed started engineers and scientists on thorough investigation of the stresses in pressure vessels and compelled them to adapt more exact methods of computation, in order to produce an economic and safe construction. Most of these investigations are scattered throughout the periodicals and transactions of engineering societies and other scientific papers and are, therefore, not readily available. The different design and safety codes which were formulated as a means of accident prevention, have remedied this situation to a certain extent. But they give rules and formulas applicable to certain materials only and presuppose a familiarity with the fundamental theories. The rapid development of new alloys and materials of construction, and some special cases, however, require a thorough knowledge of the basic problems. In this book, therefore, an attempt has been made to give for each of the different elements entering into pressure vessel design an analysis of the problems involved and a brief discussion of the methods suggested for the calculation of the stresses. Numerous references to original papers have been given to serve as a guide for further investigation of certain special problems.

The code requirements mentioned in this book are based on the 1937 and 1938 editions of the Codes for Power Boilers and Unfired Pressure Vessels. Since the methods of calculations and the rules are still being improved, it is needless to say that the designer should always consult the latest edition of the codes. However, these revisions should not materially reduce the value of this book.

A brief discussion of the principal materials of construction has been included. It gives the physical properties and fabricating characteristics of the different metals and has been written with special consideration of the needs of the designer of process equipment.

In general, it has been the intent, to give as concise a discussion as possible. For the reader who wants to make a deeper study of certain special problems, the extensive bibliography will be a convenient guide. It is hoped that this manual will become a useful reference book for the designer, process engineer and chemist.

Karl Siemon

PREFACE TO SECOND EDITION

In this edition several minor revisions were made conforming to recent revisions of the A.S.M.E. Code. In Chapter VII, the Bresse-Bryan formula, which is of more practical value, was substituted for the theoretical Bryan formula. The data on the design of copper and aluminum vessels under external pressure were augmented by Figs. 183 and 184 in the Appendix. Further, a discussion of the design of hoops was added.

K. S.

PREFACE TO THIRD EDITION

Several additions have been made to the text in this edition. In Chapter VII, a section on the permissible out-of-roundness of vessels under external pressure has been added. The discussion of Coates' method of calculating the stresses in dished heads has been augmented by the inclusion of a calculation example and of formulas applying to two-radii heads. In Chapter XI, formulas have been given for calculating the stresses in the tube sheets of heat exchangers.

The design of flanges has been modified considerably through the 1942 revision of the A.S.M.E. Code. It reflects the present thought and knowledge on flange design and gives official approval to a method that has been practiced by many designers during recent years. Some of the conceptions followed heretofore have been changed thereby. The hydrostatic end force, for example, is now considered as acting on the area bounded by the mean diameter instead of the outside diameter of the contact surface. Since it appeared inopportune at this time to rewrite the chapter completely, some minor discrepancies have crept in through the revision, which should, however, not cause any difficulties.

The charts for determining the shell thickness of monel and nickel vessels under external pressure have also been revised to agree with later minimum values of the yield point of these metals.

Karl Siemon

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PART I

MATERIALS OF CONSTRUCTION

Chapter I

SELECTION OF MATERIALS

Classification. - Materials used for the construction of pressure vessels and tanks can be divided into two general groups, metals and non-metals. The most frequently used metals are iron, copper, aluminum, nickel, lead, silver, and alloys thereof. Among the alloys particularly wide application have found the chromium-iron-nickel alloys (stainless steels), monel metal, Everdur, and the bronzes and brasses. Principal non-metals used for the construction of tanks are glass, enameled ware, stone ware, cement, rubber, wood and refractories.

Selection of Material. - In the search for a suitable material the requirements placed on the material for a certain purpose are as numerous as the applications of tanks and the emphasis on certain requirements may vary from case to case. The principal factors to be considered in the selection of a suitable material for the construction of tanks are:

1. Resistance to corrosion and chemical attack
2. Tensile strength
3. Heat resistance
4. Structure of the metal
5. Workability
6. General physical properties (heat conductivity, heat expansion, hardness, abrasion resistance, etc.)
7. Price of competitive materials

Corrosion. - Metals, as found in nature, are mostly oxides. Since the oxygen is reduced in the course of the metallurgical process of refining, the equilibrium between metal and oxygen is disturbed and the metal, therefore, always tends to revert to more stable combinations. This process may be initiated through chemical or electro-chemical attack in the presence of moisture and always involves a loss of metallic material. Under normal conditions the corrosive products consist mainly of hydroxides, but at high temperatures, simple oxides may be formed. Thus corrosion may be broadly defined as the chemical action of certain external agencies on metals which causes their deterioration or destruction.

Theory of Corrosion. - The electrochemical theory seems to explain more completely the phenomena of corrosion and is more generally applicable than any other theory which has been offered so that it is now generally accepted.

According to the fundamental electrochemical principles developed by Nernst and his pupils, iron, like all other elements, has a natural tendency to pass into solution in water. Iron, however, can enter into solution only by displacing some other element already in solution.

As every metal has a certain definite tendency to go into solution with the formation of ions, depending on the normal potential of the metal and the solution pressure, so iron has the natural tendency to pass into solution in pure water; but it will only dissolve if there is present a second substance in contact with the iron, having a lower solution pressure than iron itself, or in simpler terms, if it is able to displace some other element already in solution. In the ordinary case of iron immersed in water, this first corrosion reaction can be expressed as: $Fe(metal)+2H^{+}(ionic) = Fe^{++}(ionic)+2H(atomic)$. Through this reaction and if the hydrion concentration of the water falls below a certain value, a polarizing or insulating film of hydrogen is deposited on the surface of the metal. For the corrosion to continue, it is, therefore, necessary to destroy this film through a "depolarizer," for instance dissolved oxygen or some other oxidizing agent. This second reaction can be expressed:
$2H(atomic) + 1/2\ O_2\ (dissolved) = H_2O\ (liquid\ water)$
According to this theory, which was advocated by W. H.

Walker and his co-workers, the quantity of iron which can dissolve in water in the absence of oxygen is limited to the amount necessary to polarize the cathodic portions of the metal. In the presence of oxygen, however, the hydrogen is continuously removed from those portions so that the corrosion becomes a continuous process.

This theory still did not explain the phenomenon of pitting and the fact that corrosion often proceeds most rapidly at the bottom of deep pits and at places to which oxygen has least direct access. This gap has now been closed since the importance of the currents was recognized, which are set up by variation in the concentration of oxygen, and also, in the concentration of metallic ions, at different points on the metal. The physical phenomenon itself has been first demonstrated more than a hundred years ago. The application to the problem of corrosion is as follows: When two pieces of iron are immersed in the same liquid, and one is exposed to the action of dissolved oxygen and the other is shielded from it by means of wet rust, a current is set up, the chielded (rusty) electrode being the corroded pole or anode. Corrosion-currents can also be produced by the variation of metal-ion concentration which plays an important role in the corrosion of copper.

Electrochemical Series or Potential Series. - Every metal has a certain tendency to go into solution under simultaneous formation of ions and the force upon which this process depends is called the potential of the metal. It is determined by measuring the opposing electrical potential which must be produced to prevent the metal from going into solution. As a comparative scale, usually the so-called Hydrogen Scale of Potential is adopted, in which the potential between blackened platinum saturated with hydrogen under one atmosphere pressure and an acid solution of normal hydrogen-ion concentration (acidity) is taken as zero. The different elements have been arranged in the order of their normal solution potentials in the Electrochemical Series (see Table 1).

A more complete Electrochemical Series of the elements is: potassium, sodium, barium, strontium, calcium, magnesium, aluminum, manganese, zinc, iron, cadmium, cobalt, nickel, lead, tin, hydrogen, bismuth, copper, antimony, arsenic, mercury, silver, palladium, platinum, gold, silicon, tantalum, tellurium, carbon, boron, tungsten, molybdenum, vanadium, chromium, iodine, bromine, chlorine, fluorine, nitrogen, sulphur and oxygen.

Each element in this series is electronegative to the elements which follow and will displace them from solution under certain conditions. Thus zinc will precipitate lead from a solution of lead acetate, and

Table 1. - Solution Potentials of the Elements (The Electrochemical Series) compiled by Frank N. Speller

Element	Volume of the Element in Solution	Potential in Volts
Potassium	1	-2.9
Sodium	1	-2.7
Calcium	2	-2.6
Magnesium	2	-1.6
Manganese	2	-1.08
Zinc	2	-0.76
Iron (Ferrous)	2	-0.43
Cadmium	2	-0.40
Nickel	2	-0.22
Lead	2	-0.12
Tin	2	-0.10
Iron (Ferric)	3	-0.04
Hydrogen	1	0.00
Copper (Cupric)	2	+0.34
Antimony	3	+0.47
Copper (Cuprous)	1	+0.51
Mercury	2	+0.80
Silver	1	+0.80
Platinum	4	+0.86
Gold	3	+1.08

iron will replace copper when a steel blade is dipped in copper salt solution. The reactive metals, which are easily corrodible, appear, therefore, at the beginning and the noble not easily corrodible metals at the end of the list. However, the Electrochemical Series is not a fixed and unchangeable order, but depends on the concentration of the metal ions in the solution, on the surface conditions of metals, on the temperature, etc.

Corrosion Data and Tests. - In recent years a very voluminous literature has been published which gives information on the corrosion resistance of metals and other materials of construction (rubber, ceramics, fibrous materials, plastics, etc.) to the attack of chemicals, acid, fruit juices, organic substances, salt water, etc. In many cases, the desired knowledge on the corrosive properties of a substance may be obtained by reference to these publications. The corrosion rates are usually expressed in i.p.y., inches penetration per year, which means the metal thickness in inches removed by corrosion per year. Another method is to give the weight of the metal dissolved per unit area per day.

Where no data are available, corrosion rates may be determined by laboratory and field tests. Laboratory tests are accelerated tests which are carried out in laboratories under controlled conditions. A number of such tests have been devised which attempt to reproduce operating conditions and give a good indication of the corrosion factor. The most accurate data are obtained by field tests which are, however, more time-consuming and usually more expensive. They are carried out by placing test specimens in the operating equipment and observing the loss of metal.

REFERENCES

F. N. Speller, Corrosion Causes and Prevention. McGraw-Hill Book Company, New York, 1935.

U. R. Evans, Metallic Corrosion, Passivity and Protection. London, 1937.

Chapter II

IRON AND STEEL

Introduction. - Pure iron is difficult to prepare and nearest approaches to it are electrolytic iron (99.98% pure), Armco iron 99.9%, and high quality wrought iron 99.8%. Commercial iron always contains a larger or smaller amount of impurities. Elements which every commercial iron contains are: carbon, silicon, phosphorus, manganese, sulphur and copper. Pig iron contains:

iron	92-96%	phosphorus	0.01-4%
carbon	2.5-4.5%	manganese	0.1-2.0%
silicon	0.5-3.5%	sulphur	0.01-0.15%

Most impurities harden and strengthen the metal, but do not change the general characteristics substantially.

Commercially pure iron is a comparatively soft, tough metal of high melting point, malleable, ductile and magnetic. It is very readily forged over a wide range of temperature with exception of the range between 250° and 500° C. where it is comparatively brittle. It also welds very easily.

Chemical Properties. - Iron combines readily with oxygen, sulphur, carbon, and chlorine when heated, and decomposes steam at a red heat. It is not attacked by pure dry air at ordinary temperatures, but at 200-400° C. thin, colored films of oxides are formed and at 400° C. it begins to scale.

Iron corrodes in ordinary moist air particularly if the moisture condenses on the metal, also in impure industrial atmospheres and very rapidly in water containing dissolved oxygen, particularly if it is acid or saline, but the presence of alkalis retards attack.

Irons and steels are easily dissolved by hydrochloric acid, sulphuric acid and dilute nitric acid, but they become passive in concentrated nitric acid. Dilute alkalis do not attack iron and steel, but the action of hot concentrated caustic soda produces a very dangerous weakening, known as caustic embrittlement.

Physical Properties of Pure Iron

Melting Point	1535° C
Transformation Temperatures	900° C 1401° C
Recrystallization of Cold Worked Metal at	500-600° C
Specific Heat (20° C), cal./gm.	0.107
Thermal Conductivity, cal./cm.^{3}/1°C/sec.	0.148
Electrical Resistivity, (20°C), ohm/cm.^{3}	10.0×10^{-6}
Temperature Coeff. of Resistance, per 1°C	0.00726
Density (20°C)	7.86
Coeff of Expansion, per 1°C	11.7×10^{-6}

Typical Tensile Properties	Cold Worked	Annealed
Elastic Limit, lbs./sq. in.		13000
Yield Point, lbs./sq. in.	98000	17-22000
Tensile Strength, lbs./sq. in.	102000	32-36000
Elongation, %	3	50
Reduction of Area, %	-	80

Elastic Properties

Young's Modulus, lbs./sq. in.	26400000

(very pure commercial iron)

Elastic Limit in Compression (Annealed), lbs./sq. in.	17000
Yield Point " " " " " "	18400
Elastic Limit in Shear " " " "	12000
Yield Point " " " " " "	15000
Brinell Hardness (Annealed)	69
" " (Cold drawn 15%)	180
Scleroscope Hardness (Annealed)	20
" " (Cold worked)	37
Izod Impact Value, ft.-lb.	28

Pig Iron. - Pig Iron is produced from iron ore in the blast furnace through a reductive smelting process.

Table 2. - Classification of Pig Iron (Stoughton)

Trade Name	Total Carbon, Per Cent	Silicon, Per Cent	Sulphur, Per Cent	Phosphorus Per Cent	Manganese, Per Cent	Used for
No. 1 foundry soft	3.00	2.75-3.25	0.05 and under	0.30-1.50	0.10-1.00	Cast iron
No. 2 " "	3.25	2.25-2.75	0.05 and under	0.50-1.50	0.10-1.00	" "
No. 3 " "	3.50	1.75-2.25	0.06 and under	0.50-1.50	0.10-1.00	" "
No. 4 " "	3.75	1.25-1.75	0.065 and under	0.50-1.50	0.10-1.00	" "
Gray Forge	3.50	0.75-1.75	0.07 and under	under 1.00	0.10-1.00	In Puddling furnace
Standard Acid Bessemer Pig	3.50-4.00	1.00-1.50	under 0.08	under 0.10	[illegible]	For [illegible] in production of Bessemer steel
Basic Pig	3.50-4.00	under 1.25	under 0.08	0.10-1.50	[illegible]	In basic open-hearth furnace
Malleable Bessemer	3.50	0.75-2.00	under 0.05	under [illegible]	[illegible]	"Malleable" Iron
Silico-spiegel	1.00	5.00-15.00	under 0.05	under 0.10	[illegible]	
Ferro-silicon	0.50-2.00	10.00-50.00	under 0.04	under 0.10	[illegible]	Alloy Metals
Ferromanganese	6.00-7.00 7.00	0.50-1.00	under 0.05	....	[illegible]	
Ferro-phosphorus	1.00	1.50	under 0.05	10.00-[illegible]	....	
Spiegeleisen	4.50-6.00	1.00	under 0.04	....	[illegible]	

CAST IRON. - The qualities of cast iron are influenced by the amount and form of the carbon and by other impurities. The carbon appears as total carbon, graphite, and combined carbon. Graphite occurs in thin flakes of varying sizes which are dispersed through the metal and are composed of small component flakes which can be separated very easily. When cast iron cools slowly the carbon precipitates partly as graphite and as austenite which again later decomposes into ferrite and cementite. The carbon of cementite renders the cast iron very hard, abrasion resistant, and brittle, and is usually called "combined carbon." Slow cooling allows the precipitation of carbon in the form of graphite. Chilling or artificial rapid cooling produces a white cast iron in which the carbon appears as cementite or combined carbon.

Effect of Graphite. - Increased graphite decreases strength and shrinkage, and increases porosity, softness, and workability of the metal.

Effect of Impurities. - Silicon produces freedom from oxides and blowholes, renders iron more fluid; causes carbon to precipitate as graphite, if sulphur and manganese are kept low.

Sulphur makes the metal very tender and liable to checking at red heat, causes rapid solidification, blowholes and dirty iron, increases tendency to segregation and collection of impurities in spots. Increased sulphur increases combined carbon.

Manganese decreases machinability and counteracts the effect of sulphur.

Phosphorus reduces the melting point and renders the metal more fluid and brittle under shock at atmospheric temperature.

For iron castings several grades of pig iron are used and mixed in order to obtain the analysis which is desired. Cast iron contains more carbon (3-4%) and more impurities than steel. It is easy to melt and cast, but is not ductile and malleable and cannot be wrought to shape.

TYPES OF CAST IRON. - White iron is a cast iron whose fracture is white. It contains carbon in the form of cementite and is very hard and brittle and difficult to machine. It resists abrasion very effectively.

Mottled Iron has a white fracture with gray spots, or gray fracture with white spots.

Gray Cast Iron has a gray fracture and contains carbon in the precipitate form of graphite, which tends to decrease the strength and increase porosity and permeability to gases or liquids under pressure. It is readily machined. The strongest cast iron has about 2.75% of graphite and 1% of combined carbon (cementite). Cast Iron is produced either directly in the blast furnace or, most frequently, by remelting in the cupola, air furnace and crucible furnace and is poured into various kinds of molds such as loam-, dry-sand-, wet-sand-, clay-, and permanent molds.

Chilled Iron. If the surface of an iron casting is chilled a hard surface of white iron is produced the depth of which can be regulated at the will of the foundry-man while the interior and the part of the surface that is allowed to cool off slowly are gray. It is used where a surface of great hardness is desired that resists wear, and is backed by a metal which is stronger and not so brittle, f.i. cast-iron rolls, railroad-car wheels, anvils, etc.

Semi-Steel is a cast iron to which 25% or more of steel has been added. It is a casting of greater strength and finer grain than ordinary cast iron.

Malleable Cast Iron is cast iron which has been refined by annealing in oxides. Pig iron low in silicon, known as "malleable Bessemer," mixed with steel scrap and "return scrap," consisting of defective castings, gates, etc., are poured into sand molds. The castings then are packed in iron ore, mill scale, "bulldog," and similar forms of iron oxide, and annealed for about six to seven days at 675° to 875°C (1250° to 1600°F). Malleable cast iron has physical properties between gray iron and steel castings. Its tensile strength is 50,000 to 60,000 lbs. per sq. in. and elongation 10 to 15% in 2 in. It is used for castings with thin walls which it is difficult to produce in cast steel, f.i. wrenches, small parts for agricultural machines, for household and harness hardware, pipe fittings, etc.

Table 3. - Mechanical Properties of Cast Irons (Stoughton)

Material	Silicon per cent	Sulphur per cent	Phosphorus per cent	Manganese per cent	Tensile Strength pounds	Transverse Strength pounds
Soft iron for pulleys, small castings, good tooling.	2.20 to 2.80	Not over 0.085	Not over 0.70	0.30 to 0.70	28,000	2,200
Medium iron for engine cylinders, gears, etc.	1.40 to 2.00	Not over 0.085	Not over 0.70	0.30 to	30,000	2,500
Hard-iron cylinders, for ammonia- and air-compressors	1.20 to 1.60 * 1.2 to 1.9 **	Not over 0.095	0.70 to 0.40	Not over 0.60	25,000	2,800

* if annealed   ** if cooled fast

Densities	Specific Gravity	Weight per cubic foot
Pure Iron	7.86	490
White Cast Iron	7.60	474
Mottled Cast Iron	7.35	458
Light-Gray Cast Iron	7.20	450
Dark-Gray Cast Iron	6.80	425

Casting Temperature	1230°C (2246°F)
Shrinkage	1/8 inch per foot

WROUGHT IRON. - Wrought iron is made by refining pig iron by submitting it to the action of slags rich in iron oxide at a temperature above the melting point of the pig iron but below that of pure iron, i.e., in a pasty condition. Good wrought iron is practically pure iron with fibres of slag running through it. It is tough and reliable, resisting shock and sudden overstrain very well. It has good resistance to corrosion and to fracture from fatigue and costs about 10 to 20 per cent more than the cheapest steel. Good wrought iron has less than 0.10% of carbon and less than 2% entangled slag and other entangled oxidized substances. Commercial forms: Plates, sheets, pipe, structural shapes, bars, rivets, welding fittings, forging billets.

Uses: boilers, boiler tubes, water pipes, corrugated roofing and building sheets.

Steel. - Steel is made by purifying pig iron or scrap steel at a temperature above the melting point of pure iron, and subsequently adding a controlled amount of carbon and manganese to the melt. It is also produced by mixing pure wrought iron with material rich in carbon. It solidifies from a fluid mass into only one kind of crystal, viz., the solid solution of iron and carbon. It does not contain any slag.

The amount of carbon varies from 0.05 to 1.5%. The tensile strength increases with the carbon content up to about 0.9 per cent carbon. Above this point the tensile strength decreases again with increasing carbon content.

Working of Steel. - Mechanical work breaks up the crystals of steel, but if the working is stopped while the steel is still hot, the crystals will grow again to some extent. Steel, therefore, should be worked continuously from the highest temperature employed down to a low red heat, i.e., a temperature of about 1300° to 1400°F. Working at a blue heat, 430° to 600°F, causes brittleness after cooling.

Cold rolling produces a more accurate shape and a better surface than hot rolling but leaves the metal with internal stresses and strains, unless it is annealed afterwards. Cold pressing has a similar effect.

Table 4. - Effect of Carbon on Iron

	Carbon per cent	Tensile Strength, Annealed			Uses
		Ultimate, lbs. per sq. in.	Elastic Limit, lbs. per sq. in.	Elongation, per cent in 2 in.	
Very mild steel	0.05-0.15	40,000-48,000	24,000-30,000	44-28	Resembling wrought iron, used for cold pressing and drawing.
Mild steel	0.15-0.25	48,000-60,000	30,000-36,000	28-25	commonest variety, structural steel
Low-carbon steel	0.25-0.40	60,000-70,000	36,000-40,000	25-22	
Medium-carbon steel	0.40-0.60	70,000-80,000	40,000-48,000	22-18	strong, tough, hard; dies, rails, axles, etc.
Higher-carbon steel	0.60-0.70	80,000-94,000	48,000-56,000	18-14	
Spring steel	0.70-0.80	94,000-108,000	56,000-64,000	14-8	
Pearlitic steel	0.85	130,000	70,000	5	cutting tools.
Hypereutectoid steel	0.90-1.70	110,000-100,000	75,000-60,000	5-1	

Cast Steel. - Cast steel is poured directly from acid and basic open-hearth furnaces or from small Bessemer converters, crucibles, and electric furnaces. Carbon 0.1 to 0.7%. Shrinkage 1/4" per 1 foot. Cast steel does not flow as easily as cast iron and has to be annealed afterwards. It is used for castings which have to stand shock and abrasion: f.i. railroad switches, boiler nozzles, rings and plates of mills and stove crushers.

ALLOY STEELS. Introduction. - Alloy steels are steels which "owe their properties chiefly to the presence of an element (or elements) other than carbon," according to the definition of the International Committee upon the nomenclature of iron and steel. The elements which have proved most useful are chromium, nickel, manganese, tungsten, molybdenum, silicon, vanadium, cobalt, aluminum, etc. Most of these elements render the critical changes more sluggish. If sufficient alloying element is added the critical changes may be so far retarded that the steel becomes martensitic (air-hardening) or that even austhenitic steels are produced which preserve the structure of the iron existing at high temperatures and are entirely different from ordinary steel in all their physical properties. Alloy steels are commonly used in heat-treated condition and retain a high degree of ductility in hardened condition.

Binary Alloy Steels are composed of iron, carbon and one other alloying element, f.i. nickel steel, chrome steel.

Ternary Alloy Steels are composed of iron, carbon and two other elements, f.i. nickel-chromium steel, chrome-vanadium steel.

Effect of Various Elements.

(a) Impurities:

Sulphur. - Reduces strength and ductility and produces "red-shortness," i.e., brittleness of red-hot steel. In relatively large quantities (0.08 to 0.15 per cent) it increases the machinability of the steel.

Phosphorus. - Causes brittleness in hot and cold state. It is harmful in the case of vibrating stresses and sudden shocks.

Oxygen. - Same effect as sulphur.

Arsenic. - Raises the tensile strength, but decreases the ductility.

(b) Purposely added elements.

Carbon. - Principal hardening element. Tensile strength increases with increasing carbon up to about 0.9 per cent carbon, above this point it falls off with increasing carbon content. The melting point is lowered from 1510° to 1130°C when carbon increases from 0 to 4.2 per cent.

Silicon. - Decreases solubility of carbon in iron, in both the liquid and the solid state. Steel with 4.75 per cent of silicon has high magnetic permeability and high electrical resistance. Increases strength and ductility. Lessens acid attack.

Manganese. - Is a purifier and cleanser. High silicon content (1 to 15 per cent) produces hardness and resistance to wear.

Chrome. - Increases strength, hardness and resistance to oxidation at both high and low temperatures. Reduces scaling.

Nickel. - Increases strength, toughness and susceptibility to heat treatment; causes slight decrease of ductility. Steels with 24 per cent nickel and more are non-magnetic and have low coefficient of heat expansion.

Vanadium. - Deoxidizes iron, produces finer grain in metal and improves heat-treating qualities. Increases hardness and resistance against shock and jar. Makes steel very brittle if Mn contents exceeds 0.15 to 0.20 per cent.

Aluminum. - Very active deoxidizer. Controls grain size in alloys.

Copper. - Reduces rusting.

Tungsten. - Produces a fine, dense grain and a sharp cutting edge. In higher percentages (17 to 20 per cent) and in combination with other alloys it produces an alloy which has high strength and hardness at elevated temperatures.

Molybdenum. - Increases the yield point, endurance limits and shock resistance of steel and improves the corrosion-resisting qualities of some metals.

Crucible Steel. - Produced by melting wrought iron in clay or graphite crucibles. It is usually high in carbon: 0.35 to 1.6 per cent. Used for shafts, pistons and other parts where very high strength is required, also for high-grade tools, cutlery, etc. High in price.

Electric Steel. - Produced by melting pig iron, Bessemer steel or iron scrap under addition of ferro-alloys in electric furnaces of various types. This process is distinguished by the absence of an oxidizing, or of a carbonizing flame which makes it possible to maintain a slag rather free from oxide of iron and to purify steel to a greater extent than by other processes. It dissolves slower in dilute sulphuric and hydrochloric acid than basic open-hearth steel and is used for air-hardening and high-speed cutting tools, for parts of fine machinery and parts which require a high quality steel.

BINARY ALLOY STEELS. Silicon-Steel. - Contains up to 4 per cent Silicon. Coarse in crystals and of high electrical resistance. Used for the cores of electromagnets (4.75% Si), also for springs and bridges ("silicon structural steel," 0.50 to 1.05% Si, 0.67 to 0.95 Mn).

Manganese-Steel. - 11 to 14 per cent Mn, 1.0 to 1.3% C, 0.3 to 0.8% Si, 0.08% Ph, S very low. Very hard and abrasion resistant. Machined by grinding and often used in the form of castings. High coefficient of expansion and contraction. Low elastic limit. Used for railroad quipment, jaws of crushers, teeth of excavating machinery, liners for ball mills, burglar-proof safes, etc. Ordinarily formed by casting. Melting point 1325°C (2417°F). Tensile strength: Cast 80,000 lbs. per sq. in. and 20% elongation, rolled and heat treated 140,000 lbs. per sq. in., and 50% elongation, cold rolled 250,000 lbs. per sq. in. but little ductility. Heat treatment: Reheated to above 1,000°C (1832°F) and quenched in water.

Vanadium-Steel. - Up to 1 per cent Va. Va is a good deoxidizer of steel baths, has great strength and high elastic limit. Used for tools.

Tungsten-Steel. - Retains hardness even when heated for a few hours to 1,150°F. Used for high-speed tools and magnets.

Nickel-Steel. - High tensile strength, shearing strength (rivets), elastic limit, and resistance to fatigue and shock. Corrosion resistance increases with the percentage of nickel. Used for high-grade structural steel (2.00-3.75% Ni, and 0.25-0.35% C), automobile parts, engine forgings, railroad rails, shafting, axles, case-hardening stock for gears. Better corrosion resistance than carbon steel in the presence of the atmosphere, fresh and salt water, the ordinary acids, the smoke of locomotives, etc. Heat treatment: Quenching in water from 840°C (1,554°F) and reheating to 550°C (1,022°F). Strength, hardness, and ductility may be increased by heat treatment.

High-Speed Steels. - Average analysis: 0.60 to 0.90% carbon, 0.10 to 0.25% Silicon, 0.15 to 0.35% Molybdenum, 16 to 20% tungsten, 2 to 4% chromium, and 1% to 2% vanadium.

Air-hardening or self-hardening steel. 1.5 to 2.5% carbon, 1.5% manganese, and 5 to 8% tungsten. 1 to 2% Cr is also used in place of Mn.

Chrome Steel. - Cr increases the effect of heat treatment and the hardness of the surface. Deeper zone of hardness. Critical changes above normal temperature and slower. Combined with nickel or molybdenum it produces very tough steels. Used for rock-crushing machinery, burglar-proof safes, and ball bearings (1 to 2% Cr); armor plate (usually nickel-chrome steel), case-hardening stock (under 1% Cr), permanent-magnet steel (2 to 4% Cr).

4 to 10% Chrome Steel. - Stronger and more resistant to scaling at high temperatures and to corrosion by sulphides and oxidizing agents. Higher thermal effifiency on account of thinner wall tubing allowable. Creep strength at elevated temperatures (1,200°F) not much higher than that of carbon steel. Used in petroleum refineries for hot oil lines, heat exchangers, pressure stills, bubble tower caps, etc., particularly in the composition 4 to 6% Cr with the addition of molybdenum and tungsten.

Low Carbon Chrome Steel. - Carbon below 0.10 per cent, chromium above 11.5 per cent. Hard and corrosion resistant without heat treatment. High strength and ductility, and good working properties in heat treated condition. Fatigue resistant. Added in small quantities, silicon increases resistance to corrosion and oxidation, vanadium and tungsten improve working properties. Used for nitric acid equipment (concentrated acids), shafting used under moist conditions, parts exposed to steam, pump rods, coal screens.

Chromium Ferrites. - 16 to 18% chromium. They are easily worked hot or cold, are fairly inexpensive and have good corrosion resistance, but are not hardened by heat treatment and very little by working. Principally used for the production of nitric acid. Silicon increases resistance to cold nitric, sulphuric, citric, and oxalic acid. Copper increases resistance to salt water, hydrochloric and sulphuric acids.

High Carbon Chrome Steel.- 0.30 to 0.40% C, 11.5 to 18% Cr. Without heat treatment it is hard and attacked by some acids. In heat-treated condition it has good mechanical properties (also at elevated temperatures), hardness, and corrosion resistance (particuly if smooth surface is exposed). Used for parts of valves in high pressure steam lines, for petroleum refinery and nitric acid plant equipment, also for cutlery.

25-30 Chromium Irons. Highly resistant to oxidation at high temperatures and to oxidizing liquids and gases. However, prolonged heating to between 750° and 1,025°F, and exposition to reducing flue gases and carbon monoxide above 2,100° to 2,200°F has deteriorating effect. Manufactured as wrought steel and castings.

TERNARY ALLOY-STEELS. *Chrome-Nickel Steel.* - Usual ratio of nickel to chrome is 2-1/2 to 1. The two most common combinations are: 1.50 per cent nickel with 0.60 per cent chromium, and 3.50 per cent nickel with 1.50 per cent chromium. Carbon 0.40 to 0.55 per cent. Has very fine, dense structure, high dynamic strength and resistance to alternating stresses, vibration and shock.

Chrome-Vanadium Steel. - Similar to chrome-nickel steel in general combination of strength and ductility and principle of heat treatment. Used for springs.

Silico-Manganese Steel. - Similar to but a little more brittle than chromo-vanadium steel. Used for springs.

Chrome-Molybdenum Steel. - Has coarse grain structure and is inferior in combination of strength and ductility to the above ternary steels, but can easily be rolled, drawn into tubes, formed and welded.

Prices of Some of the Alloy Steels

(From Stoughton)

	Cents per Pound
Stainless iron	20.00
3 per cent nickel-chromium steel	6.45
3.5 per cent nickel steel	4.15
1.5 per cent nickel-chromium steel	3.60
1 per cent chrome-vanadium steel	3.60
2 per cent chromium steel	3.45
Silicomanganese steel	3.00
0.5 per cent chromium steel	2.95
Medium-manganese steel	2.90

STAINLESS STEEL ALLOYS. Introduction. - The Iron-Chromium-Nickel Alloys have a high corrosion resistance, high tensile strength and good workability. They are austenitic in structure and, therefore, cannot be heat-treated to produce definite physical properties. These properties can only be imparted by hot and cold working processes. The metal is ductile, malleable and ordinarily non-magnetic. It retains high strength at elevated temperatures and has excellent oxidation resistance up to approximately 1,600°F. The principal alloying elements, in addition to iron and small parts of impurities usually existent in iron, are chromium and nickel. Other elements sometimes added in small amounts to obtain free machining and specific corrosion resistance against certain chemicals are silicon, molybdenum, titanium, tungsten, etc.

With regard to the chromium-nickel content, the Stainless Steel Alloys may be subdivided into the 18-8 group, having 17-19% chromium and 7-9% nickel, and the 25-12 group, having 22-26% chromium and 11-13 nickel. The 18-8 alloys come in two grades; one with .08 maximum carbon, and the other with .20 maximum carbon.

The addition of nickel and chromium increases the corrosion-resistance and resistance to scaling at elevated temperatures and impact. It reduces grain growth and lessens embrittlement after long service. Higher carbon content has the effect of decreasing corrosion resistance. Titanium reduces the tendency toward carbide precipitation and intergranular corrosion. Silicon, tungsten, molybdenum tend to increase the high-temperature strength and resistance to intergranular corrosion. Molybdenum also improves the machining properties of the 18-8 types of stainless steels.

However, maximum corrosion resistance is obtained only when the carbon content is not too high and when all carbon is put in complete solution since the presence of undissolved carbides in the structure causes intergranular corrosion and weakens its resistance to chemical attack. This precipitation of carbides at the grain boundaries and corresponding weakening of the resistance to some corroding media takes place particularly when the 18-8 stainless is subjected to temperatures in the range of 800° to 1,500°F. The magnitude of this change depends upon the length of time that it is held in the above temperature range, and upon the composition. The dissolution of the carbon can be accomplished by a proper heat treatment.

18-8 Stainless Steel. - Commercial Forms. Chrome nickel steel alloys are produced in all the usual commercial forms, such as: slabs, plates, sheets, rounds, squares, hexagons, flats, forgings, castings, wire, rivets, bolts, nuts, cold rolled strip, seamless and welded tubes.

They are marketed by their manufacturers under different trade-names, f.i. Allegheny Metal, Armco Stainless 18-8, Enduro, U.S.S. Stainless, KA_2 and KA_2S, Nirosta, Silcrome, etc.

Typical Analysis of 18-8 Alloys

Chromium	17.0% - 19%
Nickel	7.0% - 9.0%
Carbon	.08 - 2.0%
Silicon	.75% max.
Manganese	.60% max.
Sulphur	.030% max.
Phosphorus	.030% max.

Physical Properties

Specific Gravity	7.93
Weight, lbs. per cu. in.	0.290
Thermal Conductivity, cal./cm.^{3}/sec./$^\circ$C	0.05
Coefficient of Linear Expansion, per $^\circ$C	
20° to 100°C	0.0000171
20° to 500°C	0.0000197
Specific Heat, cal./gm./$^\circ$C	0.117

Mechanical Properties

	Average values
Yield Point, lbs. per sq. in.	45,000
Ultimate Strength, lbs. per sq. in.	90,000
Elongation in 2 inches, per cent	60
Rockwell B Hardness	80
Olsen Cup Test, inches	.475 - .550
Elastic Modulus	29,000,000
Scaling Temperature	850°C 1,550°F
Melting Point approximately	2,560°F
Annealing Temperature	1,900° - 2,000°F

Type Numbers. - The American Iron and Steel Institute identifies the different types of stainless steel by numbers, which are given in Table A-36 (page 276).

Stainless Clad Steels. - 18-8 Stainless Steel is also available in the form of stainless clad sheets and tubes. The sheets and plates consist of ordinary soft steel sheets covered with a lining of 18-8 stainless steel, amounting to 10-20% of the total thickness of the piece. These sheets and plates are produced by hot rolling the two metals together so that a secure, tight bond is developed. The metal may be bent, formed, and otherwise worked in a manner similar to solid sheets, and is also successfully welded by either the electric or oxy-acetylene processes. Sheets are available in thicknesses No. 14 U.S.S. Gauge and heavier. Plates can be furnished up to about 1" in thickness.

Stainless Clad Tubing. - The Stainless Clad Tubing is formed from Allegheny stainless cold rolled strip with a tubular insert for strength. The inserts are open seam tubes power pressed into the lock joint stainless tubes so that the edges of the open seam contact and support the lock joint on both sides. This combination makes the finished tube approximately equivalent in strength and rigidity to commercial seamless steel tubing. The tubes may be formed and bent and cost less than solid stainless tubes.

Fabrication. - 18-8 Stainless is extraordinarily tough and possesses surprising ductility. It can be bent, spun, deep drawn, punched, threaded, forged, or otherwise deformed in the cold state. It can be polished to a mirror-like surface and is machinable with some difficulty. The metal can be soft or silver soldered and welded by the gas and arc welding processes. When welding, it is preferable to use material low in carbon, especially when particular corrosive agents will come in contact with the material. Gas welding is better suited to the joining of light gages, where it does not have opportunity to impart excessive heat to the metal. Electric welding is generally more satisfactory than gas welding because of the danger of carbon pick-up from the gas flame. Brazing is not recommended in general.

25-12 Stainless Steel. - The 25-12 Stainless Steel is a metal designed primarily for use at elevated temperatures. It retains a large percentage of its strength at elevated temperatures and resists progressive scaling at temperatures up to about 2,000°F, if not exposed to gases or chemicals which will attack the metal.

This alloy is extremely tough and ductile and can be bent, formed and welded without difficulty. It is produced in sheets and light plates up to 3/16" thick.

Typical Analysis

Carbon	.20% max.	Sulphur	.025% max.
Chromium	20.0 - 25.0% max.	Silicon	.50% max.
Nickel	10.0 - 13.0%	Phosphorus	.025% max.
Manganese	1.25% max.		

Mechanical Properties (Annealed at 2,000°F)

Tensile Strength	100,250 lbs. per sq. in.
Yield Point	49,370 " " " "
Elongation in 2"	58.0%
Brinell Hardness	149

HEAT TREATMENT. Introduction. - When a piece of steel or iron is heated above a certain temperature, which varies with the analysis of the steel, austenite forms. The temperature at which this change takes place is called the critical point and for a carbon steel with 1% C it is about 740°C. If the metal is allowed to cool again slowly a reversal of this change takes place, although in this case the critical temperature is somewhat lower (about 700°C at 1% C content). Both limits are changed considerably by a content of Cr, W, Ni, etc.

If iron is heated to the critical temperature or better above it for sufficient time and cooled slowly and evenly afterwards, a very fine even grain structure can be produced throughout and all inside stresses can be relieved which may have been set up by uneven heating or machining. These changes which occur in a piece of steel due to heating or cooling do not take place instantaneously and a certain period of time is necessary for the transformation of the characteristics of the metal, on either a rising or a falling temperature. However, if the steel is cooled by quenching, or some other method, more rapidly than the changes can reverse themselves, some of the imperfectly transformed austenite, i.e., martensite, or troostite is retained, and the iron is hardened, if the C content is at least 0.25%.

Overheating. - If iron is heated above 900°C (1,652°F) grains grow rapidly to large size and the iron becomes brittle and weak. It is "overheated." By a suitable heat treatment (repeated heating and cooling), followed by a careful working of the metal, the effect of overheating can be cured and the fine crystal structure can be "restored."

Hardening of Steel. - The steel is heated to a temperature a little above the critical point (bright-red heat) and then rapidly cooled. The suitable heat has to be determined for each steel and should not be above the point where the steel loses its magnetism.

Quenching. - The hardness, that is obtained through the hardening process, depends on the temperature to which the steel has been heated, on the speed of the cooling process, and on the temperature to which it is cooled down. Media used for quenching are ice water, ice brine, ice-sodium chloride solution, etc., for deep cooling; water, at room temperature or preheated to 40°C, saturated solutions of certain salts preheated up to 80°C; Mineral or vegetable oils, kerosene, glycerine, molten lead or solder; air current; sand.

Tempering. - After quenching, particularly in water, steel is so brittle that it is unsuitable for many applications. Very often also such tremendous strains are developed that the steel breaks under the least percussion. It is, therefore, necessary to restore the ductility and remove a portion of the hardness. This is accomplished by reheating the piece of hardened steel again in a furnace, or preferably, by immersion in oil, molten lead, molten salt baths, etc. A similar hardening effect can be produced by heating steel to above the critical range and quenching it in the first instance in some slow cooling liquid, such as cylinder oil, or molten solder at 200°C (392°F).

Annealing is applied for three principal reasons:

1. To bring the steel to a condition where it can be more rapidly machined to the desired shape and size.
2. To relieve the strains which have been set up in the steel by the rolling or forging operation.
3. To refine the grain of the metal and soften it after hardening.

In annealing steel the temperature is slowly raised to a point a little above the critical temperature and then reduced at a very slow rate. Air has to be excluded as much as possible in order to avoid undue formation of scale on the surface, because the oxygen in the scale attacks the carbon in the steel and thus produces a decarbonized surface. For stress-relief after mechanical treatment or hardening, heating to temperatures from 200° to 500°C (930°F) is sufficient.

Normalizing. - This is accomplished by heating the steel to a point above the hardening temperature and cooling it in air. Steel so treated will not be as soft as if it had been annealed but rolling and forging strains will have been removed and the coarse grain structure produced by high forging and rolling temperatures will be broken up. The removal of strains by normalizing is very helpful in preventing warpage during subsequent machining operations.

Case Hardening. - This process is applied in order to harden the steel on the surface, and is based on the slow absorption of carbon by iron or steel at a bright-red heat. In the Case Carburizing Process, the steel is packed in a metal container and surrounded by some carbonaceous material (charcoal, powdered bone, animal black, lampblack, graphite, etc.). The boxes are then raised to a temperature of about 900°C (1,650°F) and held for a certain length of time, during which the surface of the steel will absorb carbon from the surrounding compound. When such a piece is subsequently heated and quenched, the high carbon steel on the surface will become hard and the core will remain soft.

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Chapter III

COPPER, BRASSES AND BRONZES

COPPER. Introduction. - Copper is one of the most widely used metals in chemical and processing plants. It is a very durable metal and is chemically inert in the presence of water, beer, etc. at all temperatures. It dissolves very little in non-oxydizing acids, but is readily attacked in presence of oxydizing agents. Copper has a higher thermal conductivity than any other commercial metal and, therefore, provides the highest possible efficiency in operations involving the transfer of heat. The initial cost of copper is in most cases considerably less than the cost of other materials which might be used satisfactorily.

Commercial copper usually contains some impurities, such as lead, arsenic, bismuth, zinc, nickel, and silver. Small amounts of bismuth (as little as 0.05%) render copper brittle; arsenic and nickel (up to 0.5%) increase the strength, but reduce the electrical conductivity. The Arsenical Copper contains from 0.2 - 0.5% arsenic and, on account of its greater strength at higher temperatures and its greater resistance to the action of gases of combustion, is used for firebox plates.

Electrolytic and Lake Copper contains in the average about 99.88% copper (including silver). Deoxidized copper is a special product from which all cuprous oxide inclusions are eliminated. It is more highly resistant to corrosion than ordinary copper.

Commercial Forms. - Copper is produced in all usual forms, f.i. wire, sheets, strips, bars, plate, seamless tubes and pipes, angles, etc., and in most fabricated forms, like wire cloth, bolts, nuts, rivets, etc. Copper sheets, plates, segments, circles and semi-circles, up to but not exceeding 6,000 lbs. finished weight per piece, can be furnished and these sheets can be supplied in the following dimensions:

1/4" and thicker - up to 138" wide and 240" long.
1/8" to 1/4" in thickness - up to 132" wide and 240" long.
1/16" to 1/8" in thickness - up to 108" wide and 180" long.

Gauges. - The adoption of the micrometer caliper to determine the thickness of metal or the size of wire in decimal parts of an inch, and the abolition of all gauge numbers when ordering, is strongly recommended. The American or Brown and Sharpe's Gauge is applied to brass, bronze, nickel, silver and copper in the form of sheets, wire, rods and brazed tubes. Stub's or Birmingham Gauge is the standard for specifying seamless tubes.

Physical Properties of Copper

Properties	Value
Density, lbs. per cu. in.	0.322
Melting point	1981°F
Thermal Expansion between 25°-100°C	0.0000168
between 25°-300°C	0.0000178
Pattern Shrinkage 1/4" per ft.	
Mean specific heat (25°C), cal./gram/°C.	0.0917
Heat conductivity, Cal./sq.cm./cm./sec./°C.	0.934

	Annealed	Cold Rolled or drawn
Hardness Brinell (500 kg)	30-40	60-100
" Scleroscope (Universal Hammer)	6-8	20-25
Elongation, soft, % in 2 in.		57
Modulus of elasticity, annealed, lb. per sq. in.		15,000,000
Yield Point, lb. per sq. in., soft		12,000
Electrical resistivity (20°C), Microhm/cm.^{3}		1.724
Temperature coefficient (20°C), 0.0039 per °C		

Tensile Properties

	Annealed	Cold Rolled or Drawn	Cast
Tensile Strength, lb. per sq. in.	30,000- 40,000	50,000- 70,000	20,000- 30,000
Elongation in 2 in.	40-60%	2-4%	25-45%
Reduction of area	40-60%	2-4%	

Fabrication. - Copper is one of the most ductile metals and can be easily fabricated into complicated forms. It can be rolled and formed hot or cold, drawn, stamped, or spun cold. It can be joined by brazing, soldering, and the electric arc or oxy-acetylene method. It is hardened by rolling or hammering and becomes soft again, when heated to 400° - 600°C. It is also hardened by alloying with it small amounts of tin, zinc, silicon, manganese, etc.

Deoxidized sheets can be welded by either the oxy-acetylene or electric process to give the joint the full strength of annealed copper.

Several fabricators of brewing equipment have perfected methods for welding electrolytic copper which produce joints approaching the parent metal in strength.

BRASSES AND BRONZES. Introduction. - Alloys of copper and zinc are generally called brass, while alloys of copper and tin, silicon, or aluminum are called bronze. Brass as well as bronze is manufactured in cast and in wrought form and in a great variety of composition. With the variation in copper content, the different mixtures vary in color from a copper red to yellow as the copper content decreases. This variation in composition also affects the strength, ductility and hardness. Brass can be made soft and ductile by a suitable annealing treatment and can be hardened by rolling or drawing. It can be made free-cutting by adding a small amount of lead. Lead, however, has also the tendency to make brass brittle and, therefore, is undesirable in brasses used for extra deep drawing and for springs. A selection of standard brasses, which are of particular interest to the designer of process equipment, is compiled in the following list.

Classification of Brass Mill Alloys
(Whitehead Metal Products Co., New York, N.Y.)

Yellow (High) Brass. - Contains about 65% Copper, 0.6% Iron maximum, .35% Lead maximum and the balance Zinc. Bright yellow, very ductile. Good for ordinary drawing operations.

Extra Quality High Brass. - Contains approximately 67% Copper, .10% Lead maximum, .06% Iron maximum, and balance Zinc. Bright yellow, very ductile and made from best quality raw materials. Recommended for unusually deep drawing, for spinning, and for good quality springs.

Best Quality High Brass. - This is sometimes known as cartridge Brass and contains approximately 68% Copper, .06% Lead maximum, .05% Iron maximum and balance Zinc. Bright yellow, very ductile. Used for articles which require exceptionally deep drawing, such as eyelets. Also for cartridge cases, which must withstand the effects of shock.

Red (Low) Brass, 80%. - Contains approximately 80% Copper and 20% Zinc. Light golden color, very ductile. Used in the manufacture of band instruments and for wire cloth for paper mills, also where its golden color is desirable.

Red Brass, 85% (Rich Low). - Contains approximately 85% Copper and 15% Zinc. Fine golden color, very ductile. Used for imitation gold jewelry and for articles where a golden color is desirable.

Commercial Bronze, 90%. - Contains about 90% Copper and 10% Zinc. Red golden color, very ductile. Used for bullet jackets, also for articles exposed to the weather. Resist corrosion and season cracking better than ordinary brass.

Commercial Bronze, 95%. - Formerly known as gilding metal.

Nickel Silver. - Formerly called German Silver, is a brass to which a certain amount of Nickel has been added. Nickel whitens brass--the more nickel, the whiter the alloy. It is composed of about 60 to 65% Copper, 5 to 30% Nickel and the balance of the mixture Zinc, depending upon the quality designated. It is not quite so soft and ductile as brass. Yet, it may be worked with tools and spun in practically the same manner. The grain is of a very fine and dense structure enabling the metal to take and retain a high polish under varied conditions.

Yellow (Muntz) Metal. - Can be had in such forms as rods, tube sheets, circles, segments, bolts, rails, etc. It can be forged, stamped and dropforged and when rolled hot possesses greater strength, toughness and texture than mild steel. Its constituents are: Copper 60%, Zinc 40%.

Manganese Bronze. - An alloy of Copper, Tin, Zinc deoxidized by means of manganese, is readily rolled or forged at a hot heat, producing a tough, dense and close grained metal. Shafts, pump rods, bolts and nuts, valve stems and axles made from this material are of high value from an engineering standpoint because of resistance to vibration, sudden stress and shock.

Naval Bronze. - Is made to comply with U.S. Navy specifications, and has many of the characteristics of Tobin Bronze. Frequently used for making bolts, studs, nuts, turnbuckles, air pump and condenser parts, sheets, supporting plates, etc., because of its resistance to the action of salt water.

67% Brass Pipe. - 67% copper, lasting service where normal conditions prevail, where water has low hardness, conforms to Government Grade "B".

85% Red Brass Pipe. - Highest quality corrosion resistant water pipe obtainable at moderate cost. 85% Copper. Grade "A" water pipe.

SPECIAL BRONZES. Ambrac Metal. - Ambrac is a corrosion resisting white metal composed of about 65% Copper, 20% Nickel, 5% Zinc and manufactured by The

American Brass Company. It can be drawn, spun, stamped or double seamed with ease. When annealed, it has a tensile strength of approximately 55,000 lbs. per square inch and an elongation of about 25% in two inches. The tensile strength can be increased to 110,000 lbs. per sq. in. by cold working. The elongation would be correspondingly reduced to about 2% in two inches. Ambrac withstands the action of alkalis, hot gases, dilute acids, and saline solutions, and has been used successfully for laundery machinery, wine screens, salt works tubes, and condenser tubes operating at high temperatures.

Ambraloy (Am. Brass Co.). - Aluminum-Brass Alloy, responds to precipitation hardening or heat treatment, used for condenser tubes.

Tobin Bronze. - (Am. Brass Co.) Combining high resistance to salt water corrosion with light weight, ductility and torsional strength comparable to steel, Tobin Bronze is an ideal metal for marine duty. Used for hull plates, propeller shafting, fin keels and other underwater parts. Its approximate composition is 59 to 62% copper, 1/2 to 1-1/2% tin, and the balance zinc.

Everdur. - Everdur Metal is a copper-rich alloy of copper, silicon, and other elements, manufactured by the American Brass Company. It has a tensile strength approaching that of steel and at the same time retains the non-rusting and corrosion resistant properties of copper. Everdur is manufactured in four standard alloys: Everdur 1010, principal use hot rolled and annealed plates for pressure vessels; Everdur 1015, for severe cold working, rods, bolts and pipes; Everdur 1012, contains lead, used in rod form for screw machine products; Everdur 1000, casting alloy.

Everdur is very resistant to corrosive gases and saline fogs, and to a large number of normally corrosive solutions and compounds. It has excellent machining and working characteristics and can be fabricated into a variety of forms and shapes. It is also welded readily by the oxy-acetylene and by the electric-arc welding process. When remelted and cast, sound castings of high strength, toughness and workability are produced.

Physical Properties

Melting point: wrought Everdur (1010) 1019°C - 1866°F
casting ingots (1000) 1000°C - 1832°F

Density: castings 8.22, wrought Everdur (1010) 8.53 gm./cu.cm.

Weight per cu. in.: castings 0.297 lbs., wrought Everdur 0.308 lbs.

Shrinkage (same as for copper and brass) 3/16" per ft.

Thermal Conductivity per 1°F temperature difference: 19 B.T.U./Sq.Ft./Hr./1 ft. Thickness, at 68°F

Electrical Resistivity, (Ohms per Cir. Mil-foot at 20°C): Annealed 160, Hard drawn 173.

Electrical Conductivity:
(International Annealed Copper Standard at 20°C):
Copper 100%
Everdur annealed 6.5%
Hard drawn 6.0%

Coefficient of Linear Expansion 0.0000180 per 1°C at 25-300°C.

Annealed Everdur 1010 plates have the following strength values:

Modulus of Elasticity	15,000,000 lbs. per sq. in.
Tensile Strength, min.	50,000 lbs. " " "
Yield Strength,* min.	18,000 lbs. " " "
Elongation, min.	40% in 2 inches.

Everdur is non-magnetic.

*At 0.5% elongation under load.

Everdur Pipe and Tubes. Seamless, cold drawn pipe and tubes are supplied in all sizes up to 12" in diameter and in lengths up to 20 feet, depending upon diameter and gauge required.

Everdur Sheets and Plates. Sheets, plates, circles, strips, and rolls are supplied hot rolled, cold rolled, or cold rolled and annealed. Hot rolled Everdur tank plates can be furnished up to 132 inches wide and 228 inches long, depending on the thickness required. Standard flanged and dished heads and elliptical heads are also available.

Welding. Everdur can be welded by either the oxy-acetylene or electric arc process. Plates for butt welding 1/4" or thicker should be beveled. Everdur can be joined to steel.

Flux: fused borax is satisfactory. 90% fused borax and 10% sodium is better.

Brazing. Everdur Metal may be brazed and soldered in the same manner and with the same equipment as used for copper and bronze. The same fluxes which are used for welding serve for brazing too.

Herculoy - is a silicon bronze, composed of about 94.75% copper, 3.25% silicon, 0.5% tin and is manufactured by the Revere Copper and Brass Company. It has a rich golden color when finished, a strength comparable to that of the low and medium carbon steels and a high resistance to corrosion. It is non-magnetic, ductile, and can be welded by the gas or arc methods and worked hot easily.

Commercial Forms. - Herculoy is available in the form of hot or cold rolled resp. drawn sheets, plates, strips, rods, bars, and shafting; welding rods, and in ingot form for sand castings.

Physical Properties

Material	Tensile Strength (lbs. per sq. in.)	Elongation (in 2 inches)
Hot rolled sheets	68,000	51%
Cold rolled sheets	85,000	30%
Cold rolled sheets, light annealed	75,000	56%
Cold rolled sheets, full annealed	60,000	80%
Hot rolled rod-from a minimum of	67,000	60%
Cold Drawn-to a maximum of	120,000	3%
Sand cast Herculoy	53,000	28%

Density

Material	lbs. per cu. in.
Sheets	.309
Rods	.309
Ingots	.307

Specific Gravity - 8.54
Pattern Shrinkage - 3/16" per ft.
Forging Temperature - 1450°F
Coefficient of Thermal Expansion per °C (mean 0° - 100°C) - 0.0000170
Thermal Conductivity, gram-calories per second per sq. cm. per cm. per °C between 25 and 125°C--0.050
Melting Point °C--1023.5
Maximum allowable working temperature under load--600°F

Uses:

Hot water storage tanks
High pressure tanks
Household Range Boilers
Kettles, boilers and mixing bowls
Screen plates for process industries tanks and baskets
Chemical and sewage plant gates
Marine construction and hardware
Deep drawing and stamping

Chapter IV

ALUMINUM AND ALUMINUM ALLOYS

ALUMINUM. Introduction. - Aluminum is one of the lightest of all useful metals. It is soft, ductile, malleable, easily worked, and of a white color. It is, in general, resistant to water and atmospheric corrosion, also nitric acid at low temperatures, but is attacked by strong acids and alkalies. Commercial aluminum contains small amounts of impurities, consisting principally of iron, silicon and oxygen. The presence of these small amounts of impurities has a marked effect on its strength and hardness. Further additions of other alloying elements result in further increase of the strength amounting to more than 50%, but also in a lower ductility. Many of these alloys have excellent casting characteristics in contrast with pure aluminum which is rather difficult to cast. Cold rolling and working also hardens the metal and increases its strength. By a combination of alloying, strain hardening, and heat treatment, alloys with a tensile strength approaching that of structural steel have been developed. Their ductility is somewhat lower but still sufficiently high to permit a considerable amount of forming.

Commercial Forms. - Aluminum is available in all usual forms, in the form of plate, sheet, foil, bar, rod and wire, tubing, pipe, structural shapes, moldings, rivets, screw machine products, forgings, pipe fittings, and all types of castings.

Fabrication. - Commercial aluminum is comparatively soft and easily worked, rolled, drawn, spun, extruded, stamped, etc.; soldering offers certain difficulties because solder does not alloy with aluminum at a low temperature and because no suitable flux has been discovered yet which dissolves the oxide film, instantly formed on the surface of aluminum when it is exposed to the air.

Torch welding is applicable both to the manufacture of articles from wrought aluminum and to the repair of aluminum alloy castings. Metallic arc welding can be generally applied to welding of aluminum while the use of the carbon arc at present is confined to butt joints, either straight or corner joints and the simpler types of lap joints. For torch welding the common aluminum alloys 2 S and 3 S, ordinarily 2 S wire (Alcoa No. 1) is used, while the alloy 52 S and the heat-treatable alloys (51 S and 53 S) are welded with a rod consisting of 5% silicon and 95% aluminum (Alcoa No. 2). Good fluxes for use in aluminum welding are prepared by a number of reputable manufacturers. Aluminum Company of America markets a flux for torch welding aluminum known as "No. 22 Welding Flux."

Properties of Commercially Pure Aluminum

Density	2.71
Melting Point, °C	658
°F	1,217
Annealing Point	340 - 400°C
	650 - 750°F
Coefficient of linear expansion,	
per °C (0-100°)	0.0000235
per °F	0.000013
Specific heat (average, 18 - 100°C), gram - calories per °C.	0.218
Thermal conductivity at 18°C, gram - calories per cm^3 per 1°C per second	0.50
Specific electrical resistivity at 20°C, Microhms per cm^3	2.8
Temperature coefficient of resistivity, 20-100°C, per °C	0.0039
Hardness (cast or annealed), Brinell	25
Hardness (cast or annealed), Scleroscope	4 to 6
Tensile strength, lbs. per sq. in., cast	13,000
Tensile strength, lbs. per sq. in., wrought	12,000 to 13,000
Modulus of elasticity, in lbs. per sq. in.	10,400,000
Atomic weight (Oxygen = 16)	27.0
Weight in lbs. per cu. in.	0.0978
Pattern Shrinkage	.2031 in. per ft.

Strength of 2 SO at Elevated Temperatures

Temperature °F	Tensil Strength Lbs. per sq. in.
75	13,500
200	11,000
300	8,300
400	5,700
500	3,900
600	2,700

A factor of safety of at least 4 should be applied on the above values when using annealed material.

ALLOYS OF ALUMINUM. Wrought Alloys. - The wrought alloys of Aluminum may be divided into two classes depending on the manner in which their harder tempers are produced. One class comprises the alloys in which the strain hardening by definite amounts of cold work following the last annealing operation produces the different degrees of strength and hardness. The alloys in the other class depend primarily upon heat-treatment processes to develop their higher mechanical properties.

In the nomenclature, used by the Aluminum Company of America, the letter "S" indicates alloys used in the wrought condition, while the number designates the alloy composition. The letter following the symbol which designates the alloy composition, and separated from it by a dash, indicates the temper. The following temper designations are being used:

"O" soft or annealed temper.
"H" hard temper, produced by cold working the metal to the maximum amount which is commercially practicable for the different alloys; graduations: quarter, half and three-quarter hard (1/4 H, 1/2 H and 3/4 H resp.).
"T", fully heat-treated temper with maximum strength developed by heat treatment.
"W", "quenched" temper or temper resulting after the normal aging is complete.

Commercially pure aluminum, which contains up to one per cent of impurities (iron, silicon, oxygen), is designated by the symbol 2 S when in the wrought form. It is used for the fabrication of chemical equipment.

The alloys 2 S, 3 S, 4 S, and 52 S comprise the group in which the harder tempers are produced by strain-hardening. 2 S or 3 S are used in cases where ease of forming by drawing, spinning or stamping is of primary consideration and where requirements of high strength are secondary. The alloys 4 S and 52 S are much stronger than 3 S, have good forming qualities, and excellent resistance to corrosion and a high endurance limit; this holds true particularly for 52 S.

The heat-treated alloys 17 S, 24 S, and 25 S, 51 S, 53 S present a wide range of properties and are principally used for structural applications. Of these, 17 S is most widely used. 53 S has particularly high resistance to the corrosive action of the atmosphere.

"Alclad" is the trade mark for alloy products, manufactured by the Aluminum Company of America, which have a high corrosion resistance produced by means of a surface layer of aluminum of high purity or of a special aluminum alloy, alloyed and integral with the core. It is claimed that the coating not only protects the alloy which it covers, but by electrolytic action, prevents attack on sections of the base alloy which may be exposed by scratches, shearing, etc. Sheet and plate are available in Alclad 17 S and Alclad 24 S.

Casting Alloys. - The properties of Casting Alloys, like those of the wrought alloys, can be improved by alloying or by heat-treatment processes. The alloy composition is designated by a number, which in the case of heat-treatable alloys is followed by the letter "T" and a number to indicate the heat-treatment practice. Elements used for alloying are copper, silicon, magnesium, zinc, manganese, nickel and iron.

Table 5. - Nominal Composition of Wrought Aluminum Alloys
(Aluminum Company of America)

Alloy	Per Cent of Alloying Elements. Aluminum and Normal Impurities Constitute Remainder							
	Copper	Silicon	Manganese	Magnesium	Zinc	Nickel	Chromium	Tin
2S	....	....	....	....	....	....	....	....
3S	....	....	1.25	....	....	....	....	....
4S	....	....	1.25	1.0	....	....	....	....
14S	4.4	0.8	0.75	0.35	....	....	....	....
17S	4.0	....	0.5	0.5	....	....	....	....
A17S	2.5	....	....	0.3	....	....	....	....
B17S	3.5	....	....	0.3	....	....	....	....
18S	4.0	....	....	0.5	....	2.0	....	....
24S	4.2	....	0.5	1.5	....	....	....	....
25S	4.5	0.8	0.8	....	....	....	....	....
27S	4.5	0.8	0.8	....	....	....	....	0.05
32S	....	12.	....	1.	....	0.8	....	....
51S	....	1.0	....	0.6	....	....	....	....
A51S	....	1.0	....	0.6	....	....	0.25	....
52S	....	....	....	2.5	....	....	0.25	....
53S	....	0.7	....	1.25	....	....	0.25	....
70S	1.0	....	0.7	0.4	10.0	....	....	....

Table 6. - Approximate Composition of Aluminum Sand Casting Alloys
(Aluminum Company of America)

Alloy	Per Cent of Alloying Elements. Aluminum and Normal Impurities Constitute Remainder						
	Copper	Iron	Silicon	Zinc	Magnesium	Nickel	Manganese
12	8.0	....	....	....	....	....	....
43	....	....	5.0	....	....	....	....
47	....	....	12.5	....	....	....	....
108	4.0	....	3.0	....	....	....	....
109	12.0	....	....	....	....	....	....
112	7.5	1.2	....	1.5	....	....	....
122	10.0	1.2	....	....	0.2	....	....
142	4.0	....	....	....	1.5	2.0	....
195	4.0	....	....	....	....	....	....
212	8.0	1.0	1.2	....	....	....	....
214	....	....	....	....	3.75	....	....
216	....	....	....	....	6.0	....	....
220	....	....	....	....	10.0	....	....
A334	3.0	....	4.0	....	0.3	....	....
355	1.25	....	5.0	....	0.5	....	....
A355	1.4	....	5.0	....	0.5	0.75	0.75
356	....	....	7.0	....	0.3	....	....
645	2.5	1.5	....	11.0	....	....	....

Table 7. - Factors For Design Stresses at Elevated Temperatures
(Aluminum Company of America)

	Alloy	100°F	200°F	300°F	400°F	500°F
WROUGHT	3S-0	1.0	0.85	0.70	0.60	0.40
	3S-1/2 H	1.0	0.75	0.60	0.45	0.30
	3S-H	1.0	0.75	0.60	0.35	0.25
	4S-0	1.0	0.90	0.80	0.50	0.40
	4S-1/2H	1.0	0.90	0.65	0.30	0.25
	4S-H	1.0	0.80	0.55	0.25	0.20
	17S-T	1.0	0.90	0.60	0.30	0.15
	A17S-T	1.0	0.90	0.60	0.30	0.15
	24S-T	1.0	0.85	0.70	0.40	0.20
	25S-W	1.0	1.00	0.75	0.30	0.15
	25S-T	1.0	0.90	0.60	0.25	0.10
	51S-W	1.0	1.00	0.80	0.30	0.15
	51S-T	1.0	0.75	0.55	0.20	0.10
	52S-0	1.0	0.95	0.90	0.70	0.55
	52S-3/4 H	1.0	0.95	0.80	0.50	0.30
	53S-0	1.0	0.80	0.60	0.50	0.30
	53S-T	1.0	0.85	0.70	0.40	0.15
CAST	12	1.0	1.00	0.90	0.75	0.60
	43	1.0	0.80	0.70	0.55	0.40
	112	1.0	1.00	0.90	0.75	0.60
	122-T2	1.0	0.90	0.80	0.70	0.60
	122-T61	1.0	0.90	0.80	0.50	0.20
	142	1.0	1.00	0.90	0.85	0.30
	142-T61	1.0	0.90	0.80	0.65	0.35
	195-T4	1.0	0.95	0.85	0.45	0.30
	214	1.0	0.95	0.85	0.70	0.50
	220-T4	1.0	0.85	0.70	0.50	0.30
	355-T4	1.0	1.00	0.90	0.50	0.20
	355-T6	1.0	0.95	0.85	0.65	0.25
	355-T51	1.0	0.90	0.80	0.50	0.20
	A355-T51	1.0	0.95	0.90	0.30	0.15
	A355-T59	1.0	0.90	0.80	0.70	0.40
	356-T4	1.0	0.95	0.85	0.50	0.30

Table 8. - Typical Mechanical Properties of Wrought Aluminum Alloys
(Aluminum Company of America)

Alloy and Temper	Tension				Hardness	Shear	Fatigue
	Yield Strength (Set = 0.2%) Lbs. per Sq. In.	Ultimate Strength Lbs. per Sq. In.	Elongation Per Cent in 2 In. Sheet Specimen (1/16 Inch Thick)	Elongation Per Cent in 2 In. Round Specimen (0.505 Inch Diameter)	Brinell 500 Kg. 10 mm. Ball	Shearing Strength (Single) Lbs. per Sq. In.	Enduranc Limit Lbs. per Sq. In.
2S-0	4,000	13,000	35	45	23	9,500	5,000
2S-1/4H	13,000	15,000	12	25	28	10,000	6,000
2S-1/2H	14,000	17,000	9	20	32	11,000	7,000
2S-3/4H	17,000	20,000	6	17	38	12,000	8,000
2S-H	21,000	24,000	5	15	44	13,000	8,500
3S-0	5,000	16,000	30	40	28	11,000	7,000
3S-1/4H	15,000	18,000	10	20	35	12,000	8,000
3S-1/2H	18,000	21,000	8	16	40	14,000	9,000
3S-3/4H	21,000	25,000	5	14	47	15,000	9,500
3S-H	25,000	29,000	4	10	55	16,000	10,000
4S-0	10,000	26,000	20	25	45	16,000	14,000
4S-1/4H	25,000	31,000	6	15	55	17,000	14,500
4S-1/2H	31,000	35,000	5	10	65	19,000	15,000
4S-3/4H	35,000	39,000	3	7	73	21,000	15,500
4S-H	38,000	42,000	3	5	80	23,000	16,000
17S-0	10,000	26,000	20	22	45	18,000	11,000
17S-T	35,000	58,000	20	22	100	35,000	15,000
17S-RT	46,000	61,000	13	..	110	36,000	...
Alclad 17S-T	32,000	55,000	18	..	...	32,000	...
Alclad 17S-RT	40,000	57,000	11	..	...	32,000	...
A17S-0	8,000	22,000	24	27	38	15,000	...
A17S-T	24,000	43,000	24	27	70	25,000	13,500
B17S-0	8,000	22,000	22	25	38	15,000	...
B17S-T	25,000	48,000	22	25	80	29,000	15,000
24S-0	10,000	26,000	20	22	42	18,000	14,000
24S-T	43,000	65,000	20	22	105	40,000	14,500
24S-RT	53,000	68,000	13	..	116	41,000	...
Alclad 24S-T	40,000	60,000	18	..	...	39,000	...
Alclad 24S-RT	49,000	62,000	11	..	...	39,000	...
25S-0	10,000	26,000	20	22	45	18,000	9,000
25S-W	25,000	48,000	18	22	80	30,000	14,500
25S-T	35,000	58,000	20	22	100	35,000	15,000
27S-T	50,000	60,000	9	11	115	37,000	13,000
51S-0	6,000	16,000	30	35	28	11,000	6,500
51S-W	20,000	35,000	24	30	64	24,000	10,500
51S-T	38,000	48,000	14	16	95	30,000	10,500

Table 8. - Typical Mechanical Properties of Wrought Aluminum Alloys Continued

Alloy and Temper	Tension				Hardness	Shear	Fatigue
	Yield Strength (Set 0.2%) Lbs. per Sq. In.	Ultimate Strength Lbs. per Sq. In.	Elongation Per Cent in 2 In. Sheet Specimen (1/16 Inch Thick)	Elongation Per Cent in 2 In. Round Specimen (0.505 Inch Diameter)	Brinell 500 Kg. 10 mm. Ball	Shearing Strength (Single) Lbs. per Sq. In.	Endurance Limit Lbs. per Sq. In.
52S-O	14,000	29,000	25	30	45	18,000	17,000
52S-1/4H	26,000	34,000	12	18	62	20,000	18,000
52S-1/2H	29,000	37,000	10	14	67	21,000	19,000
52S-3/4H	34,000	39,000	8	10	74	23,000	20,000
52S-H	36,000	41,000	7	8	85	24,000	20,500
53S-O	7,000	16,000	25	35	26	11,000	7,500
53S-W	20,000	33,000	22	30	65	22,000	10,000
53S-T	32,000	38,000	14	20	80	24,000	11,000

REFERENCE

1. Alcoa Aluminum and its alloys. Aluminum Company of America.

Chapter V

NICKEL AND NICKEL ALLOYS

NICKEL. Introduction. - Pure Nickel is a white, malleable and ductile, also, somewhat magnetic metal, harder and stronger than iron, and of a high melting point. Due to the difference in ore supplies and methods of extraction, commercial nickel is divided into groups which are again subdivided into different grades. The principal subdivision is into the non-malleable grades, used for the production of alloys, nickel- and chrome-nickel-steels, catalyzers, etc., and malleable nickel, used as such in its fabricated forms.

Commercial nickel has the following average composition:

Nickel	99.20%	Manganese	0.10%
Copper	0.20	Silicon	0.10
Iron	0.40	Sulphur	0.005
Carbon	0.10		

Physical Properties of Nickel
(Data given are for Malleable Nickel)
(International Nickel Company)

Density	8.89 g/cc.
Weight per cubic inch	0.321 lbs.
Coefficient of Thermal Expansion per Centigrade	
(25°C - 100°C)	0.0000133 per °C
(25°C - 330°C)	0.0000145 per °C
(25°C - 600°C)	0.0000155 per °C
Electrical Resistivity	63 ohm per mil-foot
	10.45 microhm per $cm.^3$
Electrical Conductivity	16% of that of Copper
Coefficient of Electrical Resistivity	.0043 per °C (.0024 per °F)

Thermal Conductivity (about 1/6 that of copper)	.145 c.g.s. units
Specific Heat (20° - 1,400°C)	.130 cal./g/°C
Modulus of Elasticity (Young's)	30,000,000 lbs./sq. in.
Torsional Modulus	11,000,000 lbs./sq. in.
Melting Point	2635°F 1445°C
Pattern Shrinkage	1/4 inch per ft.
Stress Equalizing Annealing Temperature	300°C
Slight Oxidation (Tinting) at	400°C
Approximate Upper Limit for Continuous Operation in sulfur-free Oxidizing Atmosphere	900° - 1000°C
Annealing Range	650° - 815°C
Hot-Working Range	900° - 1,200°C
Pouring Temperature (Castings)	1,590° - 1,650°C

Table 9. - Mechanical Properties of Nickel
(International Nickel Company)

Properties	Hot-rolled	Cold-rolled and cold-drawn				Cast
		Annealed	Sheet Full-hard	Rod Normalized	Tubing As-Drawn	
Tensile Strength p.s.i.	65-80,000	65-85,000	100-115,000	80-115,000	[illegible]	[illegible]
Yield Point (0.2% offset) p.s.i.	20-30,000	20-30,000	90-105,000	60-90,000	[illegible]	[illegible]
Elongation (2") %	45-35	50-30	25-2	35-15	[illegible]	[illegible]
Reduction in area %	75-60	75-60	...	70-50		
Brinell (3000kg)	100-140	100-150	...	140-200		[illegible]
Rockwell "B" (1/16" ball, 100 kg)	50-77	50-83	95 min.	77-95	[illegible]	[illegible]
Endurance Limit, p.s.i.		30,000		39,500- 40,750		
Izod Impact ft. lbs.	90-105	80-85				

Commercial Forms. - Malleable nickel is available in all usual mill forms such as "full finished" (hot rolled and annealed) sheet, cold rolled sheets, hot rolled and cold drawn rods, hot rolled angles, cold rolled strip, cold drawn seamless tubing, welded tubing, wire, wire screen, bolts, screws, rivets and many other accessories.

Manufacturing Properties. - Nickel can be forged, stamped, drawn, spun and worked readily and is joined by the usual methods. In general, the practice for working nickel follows closely the practice for steel. Welding can be performed by the oxy-acetylene, carbon arc and metallic arc methods, including flash and seam welding. Plain and flux coated wire for gas and electric welding are distributed through the same channels as the metal. Nickel can also be silver soldered and soft soldered and brazed. In heating nickel plates over an open fire, special care has to be given that a sulphur free fuel is used.

Chemical Properties and Uses. - Nickel has a remarkable resistance to the action of air and water, and non-oxidizing acids, fused alkalies and of salts either fused or in aqueous solution. It resists concentrated caustic solutions very well and is less affected by ammonia and by amines than copper. It is used in numerous applications where the copper content of Nickel-Copper alloys is objectionable. The U.S. Department of Agriculture places no ban on the nickel content of foodstuffs because of the salts being non-toxic in character. It is largely used in the manufacture of dairy equipment because of this main characteristic. The necessity of protecting the products being handled from harmful metallic contamination is also an important consideration in other applications, f.i. caustic-soda production.

Nickel Clad Steel. - Nickel-Clad Steel is a bimetal, made by a hot-rolling process. A surface of pure, solid Nickel is permanently bonded to a steel base. The Nickel layer can be rolled from 10% to 20% or more of the total thickness of the plate. The steel is usually of the low-carbon flange quality, with nominal tensile strength of 55,000 pounds per square inch. The Nickel-Clad Steel plates can be bent and formed like ordinary steel.

MONEL. Introduction. - "Monel Metal" is the registered trade-name of a solid, silvery-white alloy containing approximately two-thirds nickel and one-third copper and produced by the International Nickel Company. The tensile strength compares favorably with that of mild steel and is retained to a high degree at elevated temperatures. It possesses remarkable corrosion resistance to oxidation, to practically all alkalies, to most acids and salts, as well as to organic substances. It resists abrasion and does not contaminate or discolor materials in contact with it. Monel Metal is tough, resilient and readily worked by all conventional methods of fabrication. It is slightly magnetic. It is not hardened by heat treatment.

Commercial Forms. - Monel Metal is available in all forms, cold rolled and full finished sheets, hot rolled plate, hot rolled and cold drawn rods and flats and special shapes, standard and extra strong I.P.S. pipe, cold drawn and hot rolled seamless tubing, angles, strip, wire, welding wire, castings, forgings, rivets, bolts, etc.

The term Full Finished Sheet means a hot rolled, annealed, pickled, 4 to 6 pass cold rolled, re-annealed and patent leveled sheet. Standard mill lengths for full finished sheets are: 60", 72", 84", 96", 108", 120", 132", 144", 156", 168", 172". Standard Mill Widths are: 24", 30", 36", 42", 44", 48", 50". Thickness up to 3/4". Standard Gauge: U.S. Standard.

Temper Chart for Cold Rolled Monel Metal Sheet

Temper	Hardness	
	Scleroscope	Rockwell "B" Scale
Cold rolled		
Dead Soft	16 and under	60 and under
Soft-Skin Hard	17-20	61-73
Quarter Hard	21-24	74-82
Half Hard	25-30	83-89
Three-Quarter Hard	31-35	90-93
Hard	36 min.	94 min.

Fabrication. - Monel can be bent, stamped, and cold worked in a similar manner as steel. Its high strength limits manual operations, such as spinning, bumping, and hand hammering, to easy shapes. It is hot worked readily within the temperature range of 1,900° to 2,100°F. In heating, sulphur-free fuel should be used, and a definitely reducing atmosphere is necessary to avoid excessive oxidation and scaling. Temperatures for box annealing are 1,350° to 1,500°F. Monel Metal can be machined without difficulty and is welded by both gas and electric methods. It can also be joined by soft as well as silver soldering and brazing.

Nominal Composition of Monel Metal

Nickel	67%	Manganese	to 1.0%
Copper	30%	Silicon	to 0.10%
Iron	1.4%	Carbon	to 0.15%

Physical Properties (Monel Metal)
(International Nickel Company)

Density	8.84
Weight, lbs. per cu. in.	0.319
lbs. per sq. ft., 1" thick plate	46
Melting Point	1,350°C (2,460°F)
Annealing Temperature	1,350 to 1,500°F
Coefficient of thermal expansion	
per °C (25°C - 100°C)	.000014
(25°C - 300°C)	.000015
(25°C - 600°C)	.000016
Heat Conductivity (0°C - 100°C), c.g.s. units (1/15 of that of copper)	.06
Specific Heat (20° - 400°C), cal. per gram per °C	0.127
Electrical Resistivity (at 0°C)	
microhm per cubic centimeter	42.5
ohm per mil-foot	256
Coefficient of Electrical Resistivity, per °C	.0019
per °F	.0011
Magnetic Transformation	approx. 95°C (203°F)
Modulus of Elasticity	26,000,000 lbs./sq. in.
Torsional Modulus	9,000,000 lbs./sw. in.
Pattern Shrinkage	1/4" per foot

Table 10. - Mechanical Property Ranges of Monel Metal
(International Nickel Company)

	Tensile Strength lbs. per sq. in.	Yield Point lbs. per sq. in.	Proportional Elastic Limit lbs. per sq. in.	Elongation in 2 inches per cent
Rod and bar,				
Cold drawn:				
Annealed	70,000 - 85,000	30,000 - 40,000	[illegible]	[illegible]
As drawn	85,000 - 120,000	[illegible]	[illegible]	[illegible]
Hot rolled	80,000 - 95,000	[illegible]	[illegible]	[illegible]
Forged	80,000 - 110,000	[illegible]	[illegible]	[illegible]
Wire, cold drawn:				
Annealed	70,000 - 85,000	30,000 - 40,000	[illegible]	[illegible]
No. 1 temper	95,000 - 110,000	[illegible]		[illegible]
Regular	110,000 - 140,000			[illegible]
Spring	140,000 - 170,000			[illegible]
Plate, hot rolled	80,000 - 110,000	40,000 - [illegible]		[illegible]
Sheet and Strip:				
Full-finished sheet	65,000 - 80,000	[illegible]	[illegible]	
Cold rolled				
Annealed	65,000 - 80,000	[illegible]	[illegible]	[illegible]
Full-hard sheet	100,000 - 120,000	90,000 - [illegible]		[illegible]
Full-hard strip	100,000 - 130,000	90,000 - [illegible]		[illegible]
Tubing, cold drawn:				
Annealed	65,000 - 85,000	[illegible]	[illegible]	[illegible]
As drawn	90,000 - 105,000	60,000 - 75,000		[illegible]
Casting	65,000 - 80,000	35,000 - 40,000		[illegible]

Table 11. - Hardness Ranges of Monel
(International Nickel Company)

Form and Condition	Brinell Number (3,000 kg)	Rockwell B 1/16" Ball, 100 kg
Rod, annealed	120 - 160	65 - 83
Rod, hot rolled;	130 - 170	70 - 85
Rod, forged	130 - 220	
Hard sheet, strip		98 min.
Cold drawn rod, normalized	160 - 220	85 - 98
Cast	125 - 150	65 - 80
Cast (hard - "S" Monel)	275 - 350	

Table 12. - Strength of Hot Rolled Monel Rod at Elevated Temperatures, in Short Time High Temperature Test*
(International Nickel Company)

Temperature Degrees Fahr.	Ultimate Tensile Strength lbs. per sq. in.	Limiting Creep Stress** lbs. per sq. in.	Proportional Limit lbs. per sq. in
Room	81,000		32,000
200	79,000		29,000
400	78,000		26,000
600	78,000		28,000
750	67,000	20,000	30,000
950	53,000		24,000
1,100	35,000		20,000
1,300	23,000		13,000

*Rate of Straining: 0.016 in.per min. for yield point,
0.026 in.per min. for tensile strength.
**0.1 per cent elongation in 10,000 hours.

"K" Monel. - "K" Monel is a hardenable alloy made from Monel by introducing small quantities of aluminum. "K" Monel has all the corrosion resistance of Monel, and, through heat treatment, yield strength is over 100,000 lbs. per sq. in. and a hardness over 300 Brinell may be developed. It is non-magnetic.

Commercial Forms. - "K" Monel is supplied as bars, rods and forgings--also as strip, in limited sizes and quantities.

Fabrication. - Prior to heat treatment, "K" Monel is fabricated like Monel. It can be forged in practically any shape forgeable in steel. It may be welded with the oxy-acetylene and metallic arc methods and silver soldered.

Composition. -

Nickel	66%	Silicon	.25%
Copper	29%	Manganese	.4%
Aluminum	3.3%	Carbon	.15%
Iron	.9%	Sulphur	.005%

Density: 8.58; or 0.31 lb. per cu. in.
Other physical constants same as for Monel.

Mechanical Properties of "K" Monel

Condition	Tensile Strength p.s.i.	Yield Strength (0.2 offset) p.s.i.	Ductility % Elong. in 2"
Hot rolled or forged	90 - 110,000	40 - 60,000	45 - 35
Forged	90 - 120,000	50 - 80,000	40 - 25
Hot rolled, heat treated	135 - 160,000	100 - 120,000	30 - 20
Forged, heat treated	135 - 165,000	100 - 125,000	30 - 20
Cold drawn	100 - 125,000	70 - 110,000	35 - 20
Cold drawn, heat treated	140 - 170,000	100 - 150,000	30 - 15

INCONEL. - Inconel is a nickel alloy consisting of approximately 78 to 80% nickel, 12 to 14% chromium, and 6 to 8% iron. It presents a very favorable combination of corrosion resistance, strength, and working properties. By cold working strengths up to 200,000 lbs. per sq. in. can be obtained. It is resistant to oxidation and free from intergranular deterioration when exposed to high temperatures. It can only be hardened by cold working.

<ins>Commercial Forms</ins>. - Inconel is available in the form of plates, sheet, strip, seamless tubing, rods, forgings, castings, and as Inconel-Clad Steel plate.

<ins>Fabrication</ins>. - Inconel can be bent, drawn, and worked in practically the complete range of modern working methods. It is machinable both in the rolled and cast form, although for free cutting a modified composition is used. Heating should be done with a low sulphur fuel and in a reducing atmosphere. Stress relieving temperatures are between 1,000°F and 1,400°F. Annealing is complete at 1,800°F. For pickling hot solutions of dilute nitric acid containing fluorides or chlorides may be used.

Inconel can be joined by soft and silver soldering, brazing, and by the oxy-acetylene and electric arc welding methods. Sheet metal of 18 gage and heavier can be readily arc welded. For oxy-acetylene welding a flux mixture of boric acid, borax and sodium fluoride, or the Oxweld "Cromalloy" flux should be used. Welded joints in Inconel are not subject to intergranular deterioration, and do not require heat treatment to bring out best corrosion resistance.

Physical Properties of Inconel
(International Nickel Company)

Density, grams per cubic centimeter	8.51
Weight, pounds per cubic inch	0.307
Melting point	2,540°F - 1,395°C
Specific heat	
77 - 212°F	0.11 BTU/lb./°F
25 - 100°C	0.11 cal./gm./°C
Heat Expansion	
100 - 200°F (38 - 93°C) per °F	.0000064
per °C	.0000115
100 - 1,400°F (38 - 760°C) per °F	.00000896
per °C	.0000161
Heat Conductivity (40 - 100°C) c.g.s.	0.036
Specific Heat (25 - 100°C)	0.109
Modulus of Elasticity (lbs./sq. in.)	31,000,000
Modulus in Torsion (lb./sq. in.)	11,000,000

Mechanical Properties

	Tensile Strength lbs./sq. in.	Yield Point lbs./sq. in.	Elongation % in 2 in.
Sheet and strip			
annealed	80 - 95,000	30 - 40,000	55 - 35
Rod			
Annealed	80 - 95,000	25 - 40,000	55 - 35
Cold drawn	95 - 130,000	70 - 105,000	30 - 15
Wire			
Annealed	80 - 105,000	25 - 40,000	55 - 35
Spring temper	165 - 185,000		10 - 2

REFERENCES

1. Bulletins published by the International Nickel Company, Inc., New York, N.Y.

PART II. DESIGN

Chapter VI

CYLINDRICAL VESSELS UNDER INTERNAL PRESSURE

Introduction. - In thin cylinders, i.e., in cylinders the wall thickness of which is small in comparison to the diameter, it can be safely assumed that the stresses in the wall due to the internal pressure are uniformly distributed over the whole section. In this case the tensile stress resisting rupture in the circumferential section (area of cylinder ring x unit stress S) is equal to the force exerted on the ends of the cylinder by the internal pressure (circular area x unit pressure p):

$$D\pi t S = \frac{\pi D^2}{4} p$$

or

$$t = \frac{Dp}{4S} \qquad (1)$$

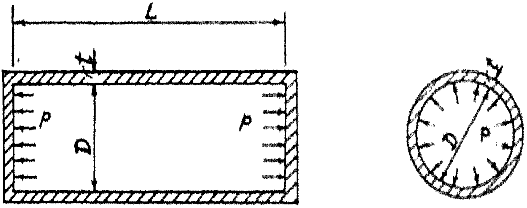

Fig. 1

In a similar way the tensile stress resisting rupture along the longitudinal section (length of cylinder x wall thickness x 2x unit stress S) is equal to the force exerted by the internal pressure on the longitudinal section of the cylinder (length of cylinder x diameter x unit pressure p):

$$2LtS = LDp$$

or $$t = \frac{Dp}{2S} \qquad (2)$$

For the calculation of the cylindrical shells these formulas are corrected to allow for corrosion, a factor of safety, and type of seam (welded or riveted).

A comparison of the two formulas shows that the resistance to rupture around the circumference is twice the resistance to longitudinal rupture; riveted pressure vessels, therefore, are usually single riveted in the circumferential seams and double or triple riveted in the longitudinal seams.

However, in the design of thick cylinders, i.e., cylinders in which the ratio of wall thickness to diameter exceeds .05 to .10, it has to be taken into consideration that the stresses in the shell are not uniformly distributed. That this is the case can be seen from the following consideration. The action of the pressure perpendicular to the inside wall tends to compress the material outward radially and to enlarge the inside circumference of the cylinder, thereby causing a circumferential or tangential tensile stress. If the walls are thick the average tangential tensile stress cannot be high, but as the internal stretching must be large at the same time on account of the high pressure, the tension along the inside circumference has to be much greater than the average across the section. The tangential stress, therefore, varies with the radius and has its greatest value at the inside surface and its minimum value at the outside surface of the cylinder.

The longitudinal or axial stress in thick cylinders caused by the action of the pressure on the ends is considered to be uniform over the section except near the ends of cylinders, where the effect of the stresses due to the head causes distortion of the shell.

Based on various assumptions as to the distribution of forces several formulas have been developed for the design of thick cylinders.

Design of Thin Cylinders. - A.S.M.E. Code. The A.S.M.E. Code for Unfired Pressure Vessels gives the following formula for the computation of thin shells under internal pressure, provided the thickness of the shell does not exceed 10 per cent of the radius.

$$t = \frac{pR}{SE} + c \qquad (3)$$

where t = minimum thickness of shell plates in weakest courses, inches.
p = maximum allowable working pressure, lbs. per sq. in.
R = inside radius of the weakest course of the shell, inches.
S = maximum allowable working stress of the shell, lbs. per sq. in.
E = efficiency of longitudinal joint.
c = allowance for corrosion, inches.

The minimum shell thickness shall be 1/8 in., for riveted construction 3/16 in. If the thickness is over 10 per cent of the radius, the outer radius is used.

The A.S.M.E. Power Boiler Code requires the following minimum thicknesses of shell plates, and dome plates after flanging:

Diameter of Shell, inches	Minimum Thickness, inches
36 or under	1/4
over 36 to 54	5/16
over 54 to 72	3/8
over 72	1/2

A.P.I. - A.S.M.E. Code. This Code uses the mean diameter of the shell in figuring the thickness, if the thickness of the shell is not exceeding 10 per cent of the inside diameter.

$$t = \frac{pD_m}{2SE} + c \qquad (4)$$

or

$$p = \frac{2SE(t - c)}{D_m} \qquad (5)$$

where t = thickness in inches.
p = maximum allowable working pressure, in lbs. per sq. in.

D_m = mean diameter in inches, before corrosion allowance is added.
S = maximum allowable working stress corresponding to the operating temperature, in lbs. per sq. in.
E = efficiency of longitudinal joint.
c = allowable for corrosion in inches.

Since the exact mean diameter is not known before the shell thickness has been calculated the following approximate formula is used which is based on the inside diameter.

$$t = \frac{pD_1}{2SE - p} + c, \tag{6}$$

or

$$p = \frac{2SE(t - c)}{D_1 + t - c} \tag{7}$$

where D_1 = inside diameter in inches before corrosion allowance is added.

The minimum thickness allowed is $\frac{D_1 + 100}{1,000}$, unless the shell is reinforced by structural members.

Design of Thick Cylinders. - Barlow's Formula:

$$t = \frac{pD_2}{2SE} \tag{8}$$

D_2 = outside diameter.

Other notations as above.

Specified by A.S.M.E. Code for Unfired Pressure Vessels, if the thickness of the shell is over 10 per cent of the radius, and for tubes. This formula assumes that the circumferential stress is uniformly distributed across the wall section and that the internal pressure acts over the outside diameter. The basic assumptions are incorrect, but the stresses computed by this formula are always on the safe side. However, wall thicknesses computed with this formula are in many cases uneconomical.

Lamé Formula. - $S = p \dfrac{D_2^2 + D_1^2}{D_2^2 - D_1^2}$ (9)

or

$$t = \frac{D_1}{2E}\left(\sqrt{\frac{S + p}{S - p}} - 1\right) + c \qquad (10)$$

for radius x:

$$S \text{ circumfer.} = \frac{pR_1^2}{R_2^2 - R_1^2}\left[1 + \frac{R_2^2}{x^2}\right] \qquad (11)$$

where D_1 = inside diameter, in.
D_2 = outside diameter, in.

Specified by the A.P.I.-A.S.M.E. Code, when the thickness of the shell is greater than 10 per cent of the inside diameter; and by the A.S.M.E. Power Boiler Code, when the shell thickness exceeds 10 per cent of the inside radius. This formula is based on the "Maximum Stress Theory" and assumes that the unit strain in the direction of the axis is constant, and that the radial and circumferential stresses are independent of the longitudinal stresses. Sections perpendicular to the axis remain, therefore, plain after deformation and all deformations are radial. The formula is applicable to cylinders with either open or closed ends. For a derivation of this formula see "Applied Elasticity" by J. Prescott.

Clavarion's Formula. -

$$S = p\,\frac{R_1^2(1 - 2\mu) + R_2^2(1 + \mu)}{R_2^2 - R_1^2} \qquad (12)$$

For $\mu = 1/3$ (steel) this formula can be written:

$$R_2 = R_1\sqrt{\frac{3S + p}{3S - 4p}} \qquad (12a)$$

where R_1 = inside radius, inches
R_2 = outside radius, inches
μ = Poisson's ratio, i.e., ratio of lateral unit deformation to lineal unit deformation within the elastic limit.

.333 for steel
.333 for wrought iron
.250 for cast iron
.333 for brass

Clavarino's formula is based on the "Maximum Strain Theory" of elastic failure which claims that failure is caused when a certain maximum strain, produced by the combined stresses, is reached. It assumes longitudinal tension as well as circumferential compression to be acting and applies to cylinders with closed ends only. For high temperatures and other cases where the value of Poisson's ratio differs from .333, the formula should be revised by inserting the correct value.

C. von Bach's Formula. -

$$S_t = \frac{m-2}{m} \quad \frac{R_1^2}{R_2^2 - R_1^2} \cdot p + \frac{m+1}{m} \quad \frac{R_2^2 R_1^2}{R_2^2 - R_1^2} \quad \frac{1}{x^2} \cdot p \qquad (13)$$

where S_t = tangential unit stress at any point distant x from the center, lbs. per sq. in.

$m = \frac{1}{\mu}$ = reciprocal of Poisson's ratio.

S_t becomes a maximum for $X = R_1$. Substituting $X = R_1$ and $m = \frac{10}{3}$ the formula reads as follows:

$$R_2 = R_1 \quad \frac{mS + (m-2)p}{mS - (m+1)p} \qquad (14)$$

$$R_2 = R_1 \sqrt{\frac{S + 0.4p}{S - 1.3p}} \qquad (14a)$$

The derivation of this formula is similar to Clavarino's formula.

Birnie's Formula. -

$$S = p \cdot \frac{D_1^2(1-\mu) + D_2^2(1+\mu)}{D_2^2 - D_1^2} \qquad (15)$$

This formula is based on the same assumptions, as Clavarino's formula except that the longitudinal stress is equal to zero. It applies only to cylinders with open ends and is used for the calculation of guns.

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Chapter VII

CYLINDRICAL VESSELS UNDER EXTERNAL PRESSURE

Introduction. - The behavior of a vessel under external pressure can best be understood by a comparison with a rod. A rod in tension can carry a load even after the yield point of the material is reached, while a rod in compression might bend and not be able to carry an increased load at stresses below the yield point, particularly if initial irregularities exist.

In a similar way, the shell of a vessel under internal pressure can carry a load even after the yield point of the material has been reached, while the shell of a vessel under external pressure might collapse at very low stresses. Initial dents and irregularities do not affect the strength of a vessel under internal pressure but weaken considerably the stability of vessels under external pressure.

The degree of stability of a vessel under external pressure depends on its length, thickness, and diameter. Relatively short, thick-walled pressure vessels are stable and fail at stresses near the yield point of the material, whereas relatively long, thin-walled pressure vessels become unstable and collapse at stresses below the proportional limit.

The calculation of a stable pressure vessel under external pressure is relatively simple. Regarding unstable pressure vessels several cases have to be considered:

(a) Short vessels: The heads are close together and tend to maintain the vessel in its original circular form. The shell collapses near the yield point of the material. The collapsing pressure depends principally on the t/D ratio. The influence of the length of the vessel or the L/D ratio is negligible.

(b) Vessels of intermediate length: As the distance between the heads increases the stiffening influence of the heads decreases. The shell collapses by unstable buckling. The collapsing pressure depends on the t/D ratio and L/D ratio.

(c) Long vessels: The heads are too far apart to influence the middle section of the shell appreciably. The shell collapses by unstable buckling. The collapsing pressure depends principally on the t/D ratio.

If a vessel is reinforced by stiffening rings of adequate rigidity the distance of these rings is considered the effective length of the vessel.

Design of Vessels Under External Pressure. Several formulas have been developed for the computation of the collapsing pressure of vessels under external pressure. The most accurate formulas are those given by von Mises. Southwell developed a theoretical formula which in its simplified form, the so-called "single hyperbola" formula, is independent of the number of lobes formed at collapse and easier to handle. Several empirical formulas have also been published. Most of these formulas are rather tedious in their application and cannot be solved for the shell thickness directly. However, they are useful in that they can serve as a check after the shell thickness has been obtained by an approximate method. A convenient procedure, which is incorporated in the A.S.M.E. Code, was developed by Saunders and Windenburg, who plotted charts for materials of certain physical characteristics giving the t/D ratio as a function of the working pressure and the L/D ratio.

Nomenclature. -

D = diameter of tube, measured to the neutral axis.
L = length of tube.
t = thickness of shell.
p = collapsing pressure.
E = modulus of elasticity.
μ = Poisson's ratio.
n = number of lobes or waves in a complete circumferential belt at the time of collapse.

Von Mises' Formula for Tubes Loaded with Radial Pressure Only (Boiler Flues). - This formula applies to thin vessels and tubes shorter than the critical length. The general corrected form (ref. 1, Eq. [B]), is:

$$p = \frac{1}{3}\left[n^2 - 1 + \frac{\lambda_1 n^4 - \lambda_2 n^2 + \lambda_3}{n^2 - 1}\right]\frac{2E}{1 - \mu^2}\,(t/D)^3 + \frac{2E\,(t/D)}{(n^2 - 1)\left[n^2\left(\frac{2L}{\pi D}\right)^2 + 1\right]^2} \tag{1a}$$

where

$$\lambda_1 = \frac{\rho\,(2 - \rho)}{(1 - \rho)^2} \qquad \lambda_2 = \rho[3 + \mu + (1 - \mu^2)\rho]$$

$$\lambda_3 = \rho\,(1 + \mu) - \rho^2\left[\mu\,(1 + 2\mu) + (1 - \mu^2)(1 - \rho\mu)\left(1 + \frac{1 + \mu}{1 - \mu}\rho\right)\right]$$

$$\rho = \frac{1}{n^2\left(\frac{2L}{\pi D}\right)^2 + 1}\;.$$

By omitting powers of ρ higher than the first, the following equation (ref. 1, Eq. [D]) is obtained:

$$p = \frac{1}{3}\left[n^2 - 1 + \frac{2n^2 - 1 - \mu}{n^2\left(\frac{2L}{\pi D}\right)^2 - 1}\right]\frac{2E}{1 - \mu^2}\,(t/D)^3 + \frac{2E\,(t/D)}{(n^2 - 1)\left[n^2\left(\frac{2L}{\pi D}\right)^2 + 1\right]^2} \tag{1b}$$

This form is furthermore reduced to the approximate equation D by neglecting the constants 1 and μ in comparison with n^2:

$$p = \frac{n^2}{3}\left[1 + \frac{2}{n^2\left(\frac{2L}{\pi D}\right)^2 - 1}\right]\frac{2E}{1 - \mu^2}\,(t/D)^3 + \frac{2E\,(t/D)}{n^2\left[n^2\left(\frac{2L}{\pi D}\right)^2 + 1\right]^2} \tag{1c}$$

In the original publication, the denominator of λ, was erroneously given as $(1 - \rho^2)$ instead of $(1 - \rho)^2$.

Therefore, the corrected version by Windenburg and Trilling (ref. 6) is given here. The average deviation from formula (1a) is less than one-half of one per cent for formula (1b) and less than 2 per cent for formula (1c). However, when L becomes infinite, formula (1c) gives values of the collapsing pressure which are 33-1/3 per cent too high.

Von Mises' Formula for Tubes Loaded with Both Radial and Axial Pressure. - This formula applies to short tubes and vessels and in its general form (ref. 2, Eq. [6]) is written:

$$p = \left[\frac{1}{3} \left\{ \left[n^2 + \left(\frac{\pi D}{2L} \right)^2 \right]^2 - 2\mu_1 n^2 + \mu_2 \right\} \frac{2E}{1 - \mu^2} (t/D)^3 + \frac{2E(t/D)}{\left[n^2 \left(\frac{2L}{\pi D} \right)^2 + 1 \right]^2} \right] \frac{1}{n^2 - 1 + \frac{1}{2} \left(\frac{\pi D}{2L} \right)^2} \quad . \qquad (2a)$$

where

$$2\mu_1 = 2 + \lambda_2 = [1 + (1 + \mu)\rho][2 + (1 - \mu)\rho]$$

$$\mu_2 = 1 + \lambda_3 = (1 - \rho\mu)[1 + (1 + 2\mu)\rho - (1 - \mu^2)(1 + \frac{1 + \mu}{1 - \mu}\rho)\rho^2]$$

ρ, λ_2, and λ_3 are defined in formula (1).

By neglecting factors of comparatively small value the following approximate equation (ref. 2, Eq. [7]) is obtained:

$$p = \left\{ \frac{1}{3} \left[n^2 + \left(\frac{\pi D}{2L} \right)^2 \right]^2 \frac{2E}{1 - \mu^2} (t/D)^3 + \frac{2E(t/D)}{\left[n^2 \left(\frac{2L}{\pi D} \right)^2 + 1 \right]^2} \right\} \frac{1}{n^2 + \frac{1}{2} \left(\frac{\pi D}{2L} \right)^2} \qquad (2b)$$

Formula (2a) can be used in all cases because the values obtained are always lower than the values obtained by formula (1a) and are, therefore, on the safe side, the average difference being 3 per cent. The approximate formula (2b) gives an average deviation of 1.5 per cent from formula (2a), but is not applicable, when L becomes infinite, for the same reason as equation (1a). All von Mises' equations are given in a notation which differs slightly from the original publication (see ref. 6).

Southwell's Formula. - Southwell obtained the following general formula for the collapse of thin tubes shorter than the critical length (ref. 3):

$$p = 2E \frac{t}{D}\left[\frac{Z}{n^4 (n^2 - 1)} \left(\frac{D}{L}\right)^4 + \frac{1}{3} \cdot \frac{m^2}{m^2 - 1} (n^2 - 1) \frac{t^2}{D^2} \right] \quad (3a)$$

where 1/m is Poisson's ratio μ, and Z is a factor depending upon the type of end constraints. Other notations are as given above.

This formula gives separate curves for successive values of n the envelope of which, Southwell showed, (ref. 4), is very closely represented by the following single hyperbola formula:

$$p = \frac{32}{9} E \frac{{}^{2}}{LD} \sqrt[4]{\frac{Z}{36}\left(\frac{m^2}{m^2 - 1}\right)^3 \frac{t^2}{D^2}} \quad (3b)$$

The constant Z for an ideal type of end constraint was evaluated by Southwell (ref. 3 and 4) and verified experimentally by Cook (ref. 11). The value obtained was

$$Z = \frac{\pi^4}{16}$$

By inserting this value of Z in equation (3a), the following general Southwell equation is obtained (ref. 3):

$$p = \frac{1}{3} (n^2 - 1) \frac{2E}{1 - \mu^2} (t/D)^3 + \frac{2E (t/D)}{(n^2 - 1) n^4 \left(\frac{2L}{\pi D}\right)^4} \quad (3c)$$

Similarly Southwell's approximate hyperbola formula becomes:

$$p = \frac{8\sqrt{6}\pi}{27} \cdot \frac{E}{(1 - \mu^2)^{3/4}} \cdot \frac{(t/D)^{5/2}}{L/D} \quad (3d)$$

Formula (3d) gives values of the collapsing pressure which are on the average 12 per cent lower than those obtained by formula (1a) and its application is restricted to tubes shorter than the critical length.

Formula of the U.S. Experimental Model Basin. - (Ref. 5 and 6)

$$p = \frac{2.42E}{(1 - \mu^2)^{3/4}} \; \frac{(t/D)^{5/2}}{\left[\frac{L}{D} - 0.45(t/D)^{1/2}\right]} \qquad (4a)$$

Substituting $\mu = 0.3$ (steel), the equation becomes

$$p \quad \frac{2.60E(t/D)^{5/2}}{\frac{L}{D} - 0.45(t/D)^{1/2}} \qquad (4b)$$

Formula (4a) is an approximation to formula (2a) and deviates on the average about one per cent therefrom. It has a similar relation to the von Mises' formula (2a) as the relation of the Southwell approximate hyperbola formula (3a) to the general formula (3b) and on account of its simplicity is recommended for the design of pressure vessels shorter than the critical length.

Bresse-Bryan Formula. This is a formula for the two-dimensional collapse of a thin tube longer than the critical length (ref. 7).

$$p = \frac{1.58E}{1 - \mu^2} \cdot (t/D)^3 \qquad (5)$$

This formula agrees with Carman's tests and eq. (11), and is applicable to t/D ratios up to 0.023 at normal temperatures. Above this limit, a tube becomes thick and fails at stresses approaching the yield point.

Stewart's and Carman's Empirical Formula. - This formula is based on tests with commercial lap-welded steel tubes (Stewart, ref. 8) and thin seamless steel tubes (Carman, ref. 9) and is similar to Bryan's formula.

$$p = 50.2 \times 10^6 \left(\frac{t}{D}\right)^3 \qquad (6)$$

where D = outside diameter.

Critical Length. - The critical length Lc of a tube is the minimum length beyond which the resistance of the tube to collapse due to external pressure is independent of the length. Southwell (ref. 3, p. 227) suggested the following formula:

$$Lc = KD\sqrt{\frac{D}{t}} \tag{7}$$

where $$K = \frac{4\sqrt{6}\pi}{27} \cdot \sqrt[4]{1 - \mu^2} = 1.11 \text{ (for } \mu = 0.3)$$

Cook (ref. 10) obtained by experiments a value of the constant

$$K = 1.73$$

for values of t/D from 0.0097 to 0.0146.

Determination of the Number of Lobes n. - The value of n to be used in the formulas is the integral value which makes p a minimum. An approximate value can be obtained by the following formula, (ref. 5, p. 214, and ref. 6):

$$n = \sqrt[4]{\frac{3/4\ \pi^2(1 - \mu^2)^{1/2}}{(L/D)^2(t/D)}} \tag{8a}$$

For $\mu = 0.3$ this formula becomes

$$n = \sqrt[4]{\frac{7.06}{(L/D)^2(t/D)}} \tag{8b}$$

Application of Formulas. - The Southwell as well as the von Mises equation cannot be solved directly for t. The most convenient method is, therefore, to calculate the thickness t by the approximate Southwell equation (3d). Then find the number of lobes n with equation (8b) and use the equation (2a) or (3a) as a check by substituting the values for t and n.

A.S.M.E. Code. - The A.S.M.E. Code for Unfired Pressure Vessels gives a chart for the design of pressure vessels subjected to external pressure which presents a series of curves of the t/D ratio in relation to the L/D ratio and working pressure. In the construction of these curves three formulas have been used corresponding to the different L/D ratios and they have been selected in such a manner that no formula was represented in a region where it was not applicable (ref. 5 and 6).

The formulas which have been applied are:

(1) For the horizontal section of the curves in the upper left-hand corner representing short thick vessels which fail when stresses in the shell reach the yield point:

$$p = 5P = \frac{2S_y\ t/D}{1.05} \tag{9}$$

where p = collapsing pressure, lbs. per sq. in.
P = working pressure, lbs. per sq. in.
S_y = yield point of the material, lbs. per sq. in.
t = shell thickness, inches.
D = diameter, inches.

(2) For the inclined central portion of the curves representing the medium long vessels which fail by unstable buckling at stresses below the yield point, but the heads of which offer additional strength to the central section of the shell:

$$p = 5P = \frac{2.60E\,(t/D)^{5/2}}{L/D - 0.45\,(t/D)^{1/2}} \tag{10}$$

where E = modulus of elasticity, lbs. per sq. in.
L = effective length of vessel, inches.

Other designations as above.

(3) For the horizontal section of the curves on the right-hand side representing long vessels where the stiffening influence of the heads is negligible:

(a) for t/d equal to or less than 0.023:

$$p = 5P = 50.2 \times 10^6\ (t/D)^3 \quad \text{(Carman's formula, ref. 9)} \tag{11}$$

(b) for t/d greater than 0.023 (thick tubes)

$$p = 5P = 86{,}670\ t/D - 1386 \quad \text{(Stewart's formula, ref. 8)} \qquad (12)$$

The chart, Fig. U-17, in the A.S.M.E. Code is constructed for carbon steel having the following physical properties:

Yield Point S_y = 27,500 lbs. per sq. in.
Proportional Limit S_p = 22,500 lbs. per sq. in
Elastic Modulus E = 29,000,000 lbs. per sq. in.
Poisson's Ratio $\mu = 0.3$
Factor of Safety = 5

Based on the same formulas charts have been plotted here for the following materials:

Chart	Material	Yield Point	Elastic Modulus
Fig. 2	Carbon Steel	27,500	29×10^6
Fig. 3	18-8 Stainless Steel	40,000	29×10^6
Fig. 4	Monel	25,000	26×10^6
Fig. 5	Nickel	15,000	30×10^6

Fig. 2 may also be used to figure the collapsing pressures of vessels constructed of steels with different physical properties. In this case the values of the pressure on the horizontal lines are multiplied with a factor a $\frac{S_y}{27{,}500}$ and the values on the inclined lines are multiplied with a factor $b = \frac{E}{29 \times 10^6}$, where S_y = yield point and E = elastic modulus of the material.

Moment of Inertia of Stiffening Rings. - A.S.M.E. Code. In the A.S.M.E. Code for Unfired Pressure Vessels a chart, Fig. U-21,* is given for determining the required moment of inertia of stiffening rings which is based on the following general formula

$$I_s = \frac{0.035}{E} \cdot D^4 \frac{L}{D}\, p \qquad (13)$$

By substituting $E = 29 \times 10^6$ and $p = 5P$ this formula becomes:

$$I_s = 6 \times 10^{-9}\, D^3\, LP \qquad (13a)$$

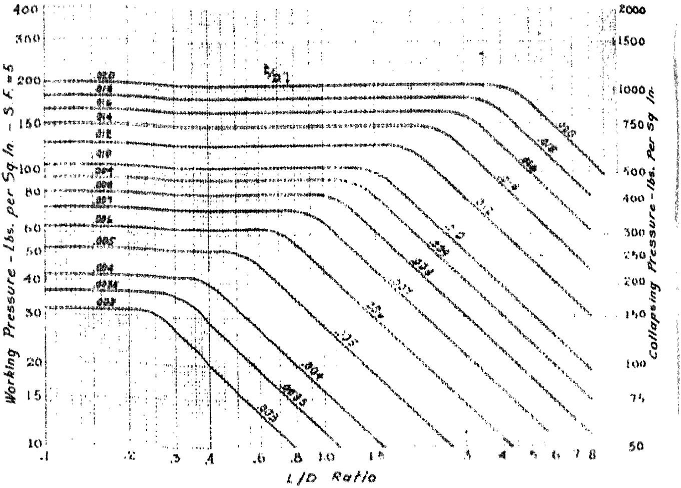

Fig. 2. Chart for Determining Shell Thickness of Cylindrical Vessels Subjected to External Pressure--Carbon Steel.

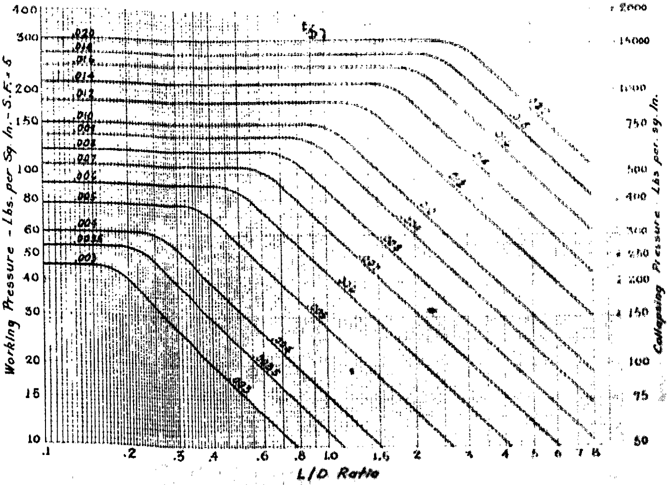

Fig. 3. Chart for Determining Shell Thickness of Cylindrical Vessels Subjected to External Pressure--18-8 Stainless Steel.

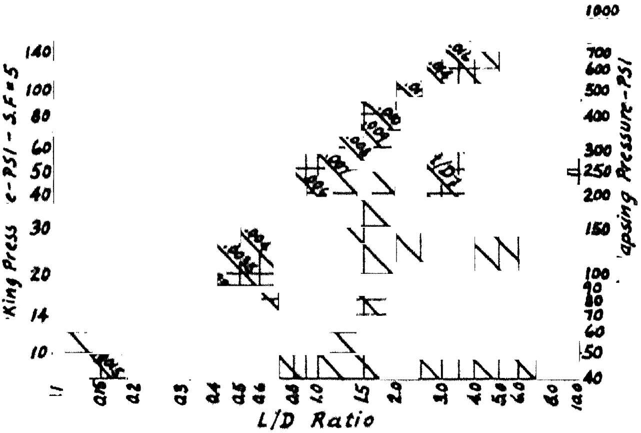

Fig. 4. Chart for Determining Shell Thickness of Cylindrical Vessels Subjected to External Pressure--Monel.

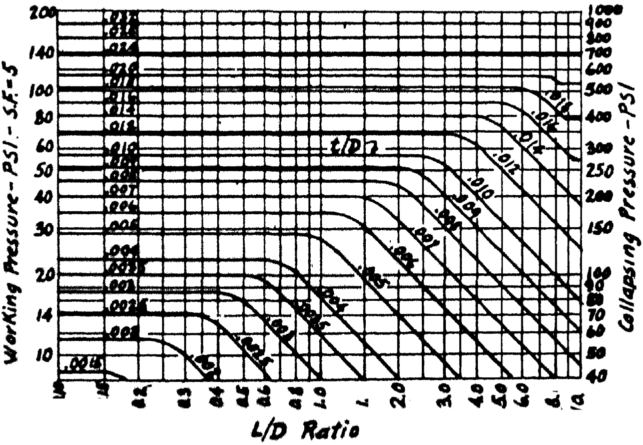

Fig. 5. Chart for Determining Shell Thickness of Cylindrical Vessels Subjected to External Pressure--Nickel.

where I_s = moment of inertia of the cross section of the stiffening ring about the centroidal axis parallel to the axis of the vessel, in.^{4}

D = outside diameter of the vessel, in.

L = center-to-center distance between stiffening rings, in.

P = working pressure, lbs. per sq. in.

p = collapsing pressure, lbs. per sq. in.

This formula has been derived from Levy's formula.

Levy's Formula. - Levy developed the following formula for the buckling of a circular ring under uniform external pressure (ref. 12 and 13):

$$q = \frac{3EI}{r^3} \tag{14}$$

where q = load per unit circumferential length of the ring.

r = radius of the neutral axis of the ring.

The collapsing strength of stiffeners is usually figured at least 10 per cent higher than the strength of the shell.

$$q = 1.1\,pL = 5.5PL \tag{15}$$

p = collapsing pressure, computed by the above formulas, lbs. per sq. in

L = center-to-center distance between stiffening rings, in.

A strip equal in width to the flange width of the stiffener is usually included in the moment of inertia.

$\frac{I_c}{I_s}$ = 1.3 to 1.7 (majority of values 1.4 to 1.5)

I_c = I of compound section (shell + stiffener)

Gaps in Stiffening Rings necessitated by the design of the vessels should not be larger than the length of 1/4 lobe which can be obtained from the von Mises equation.

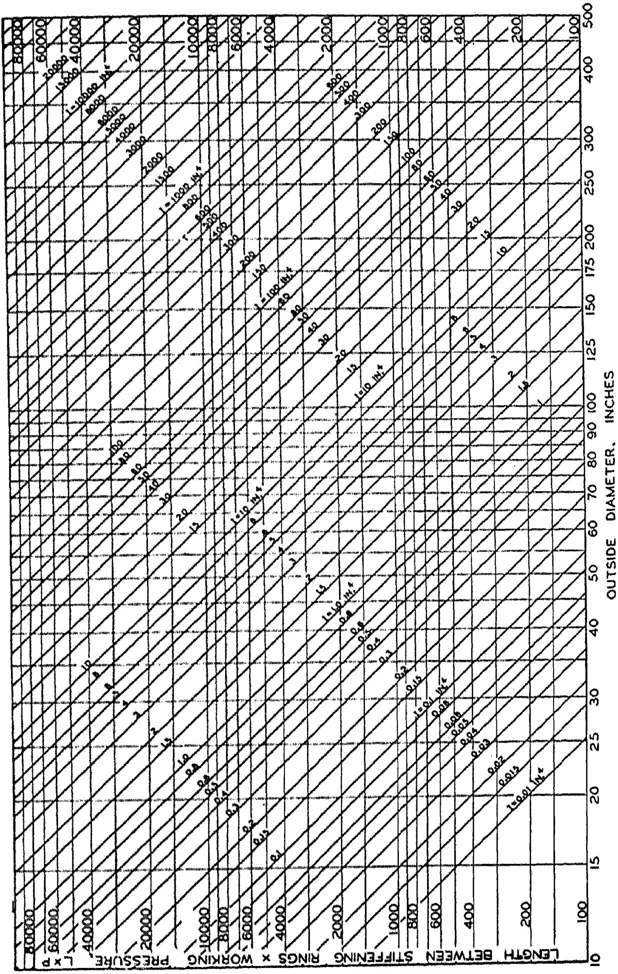

Fig. 5a. Required Moment of Inertia of Stiffening Rings for Cylindrical Vessels Subjected to External Pressure

Foeppl's Formula. - Foeppl obtained a similar formula for the buckling of a circular ring under external pressure:

$$p = \frac{3EI}{r^3L} \qquad (16)$$

where p = collapsing pressure.
I = moment of inertia of the cross section of the ring.
r = radius of the ring to the neutral axis.
L = length of the ring.

By inserting for the factor L the length of the shell between stiffeners and allowing a factor of safety of 10% this formula becomes

$$1.1\ p = \frac{3EI}{r^3L}$$

or

$$I = \frac{0.37r^3Lp}{E} \qquad (17)$$

Since a vessel under external pressure contracts to a certain extent the stiffeners should not be made stronger than necessary as the stresses in the portion of the shell adjoining the stiffeners may become too great and cause a failure.

Formula for Thick Shells. - If a buckling of the shell is not to be expected the following formula can be used:

$$R_2 = R_1 \sqrt{\frac{\mu S}{\mu S - (2\mu - 1)p}} \qquad (18)$$

where R_1 = inside diameter, in.
R_2 = outside diameter, in.
μ = Poisson's ratio.
S = allowable stress, lbs. per sq. in.
p = external pressure, lbs. per sq. in.

For $\mu = 0.3$ this equation becomes

$$R_2 = R_1 \sqrt{\frac{S}{S - 1.7p}} \qquad (18a)$$

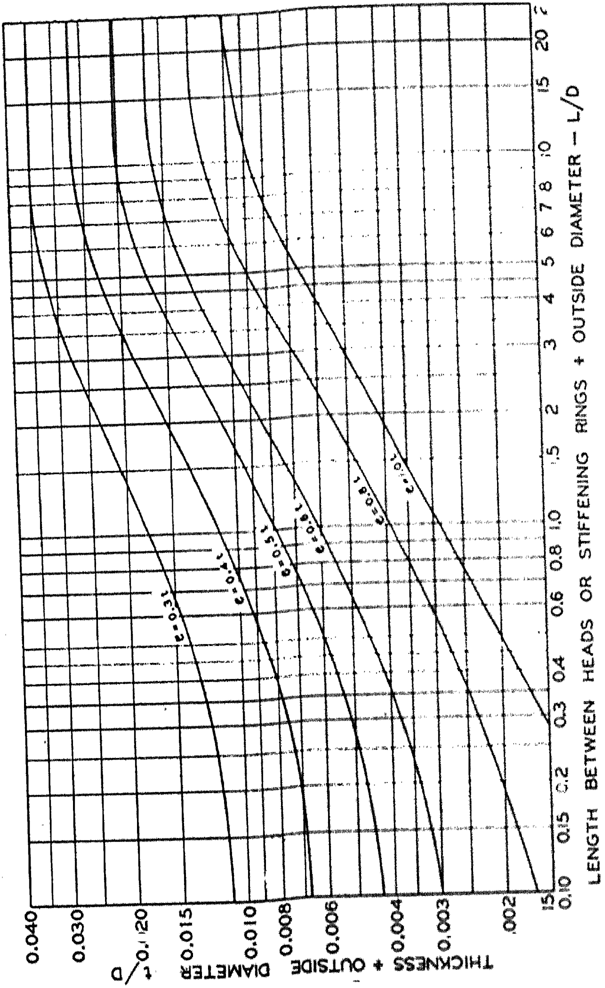

Fig. 5b. Allowable Difference in Maximum and Minimum Diameters for Cylindrical Vessels Subjected to External Pressure (Maximum Diameter – Minimum Diameter = e.)

Out-of-Roundness. - The out-of-roundess, e, is the difference between the maximum and minimum diameters in any plane perpendicular to the longitudinal axis of a vessel. Such variation from the true circular form predisposes a vessel to collapse and thereby reduces its strength. The Boiler Code Committee, therefore, has set an upper limit for the allowable out-of-roundness, which may be obtained from Fig. 5b, which has been reprinted from the Unfired Pressure Vessel Code. The curves are based on the empirical formula.

$$e/t = 0.015\ n + \frac{1.8}{n(100\ t/D)} \tag{19}$$

where n is the number of lobes in a complete circumferential belt of the shell at collapse, t is the thickness of the shell, and e is the allowable out-of-roundness. The equation is analogous to the slenderness formula of columns, the pressure vessel being considered to be made up of a series of columns, one half lobe-length long, extending end-to-end circumferentially around the vessel.

In the case of vessels having longitudinal joints of lap construction, the difference between the maximum and minimum diameters may be as great as that given by Fig. 5b, plus the thickness of the plate. Measurements shall be taken on the completed vessel in a sufficient number of planes to insure that the entire surface of the shell meets the requirements.

Example Given: Pressure vessel having t/D ratio of 0.004, and an L/D ratio of 1.5.

Required: Maximum out-of-roundness permitted.

Solution: From Fig. 5b, the allowable out-of-roundness e is found to be 0.92 t.

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Chapter VIII

HEAT STRESSES IN SHELLS

Lorenz' Formula. - If a thick cylindrical vessel and tube is subjected to heat throughput stresses are set up in the axial and circumferential directions due to the difference in temperature. The stresses are distributed over the cross section of the wall like the stresses produced in a beam through a bending moment. They change from a maximum positive value on one surface to a maximum negative value on the other surface (Fig. 6), the warmer side being compressed and the cooler side being in tension. Dr. Rudolph Lorenz (Ref. 1) developed the following formulas for the calculation of the temperature stresses:

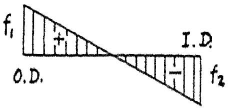

Fig. 6
Temperature Stresses

$$f_1 = \alpha \cdot G \cdot (T_1 - T_2) \cdot A \cdot B \qquad (1)$$

$$f_2 = \alpha \cdot G \cdot (T_1 - T_1) \cdot A \cdot C \qquad (2)$$

where f_1 = tangential stresses at O. D., lbs. per sq. in.
f_2 = tangential stresses at I. D., lbs. per sq. in.
T_1 = outside metal temperature, deg. F.
T_2 = inside metal temperature, deg. F.
α = linear coefficient of expansion at the working temperature.

$$G = \frac{1}{2} \cdot \frac{m}{m+1} \quad E;$$

E = elastic modulus at working temperature, lbs. per sq. in.

$$A = \frac{m+1}{m-1}$$

$$B = \frac{2}{\beta^2 - 1} - \frac{1}{\log \beta}$$

$$C = \frac{2\beta^2}{\beta^2 - 1} - \frac{1}{\log \beta}$$

$$\frac{1}{m} = \mu = \text{Poisson's Ratio}$$

$\beta = \frac{r_1}{r_2}$;

r_1 = outside radius, in.;

r_2 = inside radius, in.

Values of A corresponding to various values of m are given in Table 13, and values of B and C corresponding to various values of β are given in Table 14.

The limiting values of β are:

$\beta = 1$	$B = -1$	$+1$
	$B = 0$	$+2$

Positive values of the stresses mean compression, and negative values tension.

f_1: $-$ f_2: $+$ for T_1 larger than T_2

f_1: $+$ f_2: $-$ for T_1 smaller than T_2

The temperature drop may be calculated by the following formula:

$$T_1 - T_2 = \frac{6 \cdot Q}{k} \quad \log \beta \qquad (3)$$

where Q = heat throughput, B.t.u. per foot of length per hour,

k = thermal conductivity, B.t.u./(sq. ft.)(deg. F)(hour)(in.)

k = 150 for 18-8 Cr Ni
= 230 for 4-6 Cr
= 310 for steel

If the vessel is also subjected to internal or external pressure the resultant stresses are obtained by adding the temperature stresses to the pressure stresses. (Fig. 7.)

The Lorenz formula is based on an elastic material within the elastic limit and can, therefore, be safely applied only to relatively low temperature and to materials with high creep resistance.

Table 13. Values of A Corresponding to Values of m.

m	3	3.1	3.2	3.3	3.33	3.4
A	2	1.95	1.91	1.87	1.86	1.83
m	3.5	3.6	3.7	3.8	3.9	4
A	1.80	1.77	1.74	1.71	1.68	1.67

Table 14. Values of B and C Corresponding to Values of β.

β	-B	C	β	-B	C
1	1.0000	1.0000	2.5	0.7104	1.2896
1.02	1.0000	1.0000	2.6	.6993	1.3007
1.05	0.9839	1.0161	2.7	.6889	1.3111
1.1	0.9683	1.0317	2.8	.6788	1.3212
1.2	.9388	1.0612	2.9	.6693	1.3307
1.3	.9130	1.0870	3.0	.6602	1.3398
1.4	.8887	1.1113	3.1	.6515	1.3485
1.5	.8663	1.1337	3.2	.6431	1.3568
1.6	.8456	1.1544	3.3	.6353	1.3647
1.7	.8263	1.1737	3.4	.6278	1.3722
1.8	.8003	1.1997	3.5	.6204	1.3796
1.9	.7916	1.2084	3.6	.6135	1.3865
2.0	.7760	1.2240	3.7	.6067	1.3933
2.1	.7613	1.2387	3.8	.6003	1.3397
2.2	.7476	1.2524	3.9	.5940	1.4160
2.3	.7347	1.2656	4.0	.5880	1.4120
2.4	.7221	1.2779			

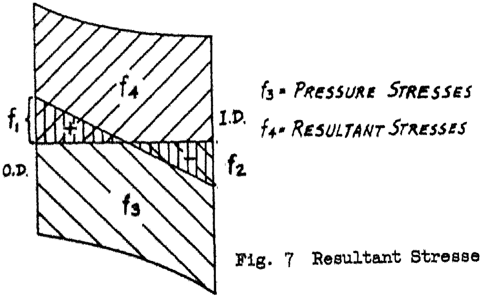

Fig. 7 Resultant Stresses

Bailey's Formula. - In the case of pressure vessels and tubes, made of material with low creep resistance and subjected to internal pressure and heat input, such as tubes in cracking coil furnaces and high pressure boilers, the elastic temperature stresses ar reduced through creep relief. R. W. Bailey (Ref. 5) developed the following formula for the hoop stress at the inside of a tube subjected to heat input and internal pressure

$$f_2 = \frac{p \cdot K^{2/N}}{A \cdot N \cdot (K^{2/N} - 1)} \tag{4}$$

where f_2 tangential stress at the inside diameter of the tube, lb. per sq. in.,
p = internal pressure, lb. per sq. in.,
K = ratio of outside diameter to inside diameter,
N = coefficient depending on the heat input Q
A = coefficient depending on the material

Some typical values of A and N are:

18-8 Cr Ni	$A = .555$	$N = 12/(1 + Q/2820)$
4-6 Cr	$A = .6$	$N = 6/(1 + Q/4320)$
Steel	$A = .6$	$N = 6/(1 + Q/5900)$

where Q = heat input in B.t.u. per ft. of length per hour.

Where the heat input, the internal pressure, and the inside metal temperature is known, the ratio K, and thereby the thickness of the tube, is obtained from the following equation

$$K^{2/N} = \frac{S_2}{S_2 - p/A \cdot N} \tag{5}$$

where S_2 = Safe working hoop stress at the inside diameter of the tube, lb. per sq. in.

The allowable working stress is usually made equal to the stress producing a creep of one-tenth of one per cent per hundred thousand hours at the respective working temperature. The highest stress and creep occur at the inside diameter of the tube.

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Chapter IX

SHELL THICKNESS OF OPEN TANKS

TANKS. Cylindrical Tanks. - The shell thickness of open round tanks is computed in a similar way as the one of cylindrical pressure vessels. The pressure existing in an open vertical tank when it is filled is equal to the pressure exerted by the liquid column and varies from zero at the top to a maximum at the bottom. Since 1 lb. per sq. in. = 2.309 ft. of water the equation for the shell thickness can be written:

$$t = \frac{2.6 \times H \times D \times G}{E \times S} + c$$

where t = required thickness of course in inches
H = height or head in feet from the bottom of the course under consideration to the top of the tank
D diameter of the tank in feet
G specific gravity of the liquid contained
S allowable unit stress for plate in pounds per square inch
E efficiency of longitudinal joint
c corrosion allowance

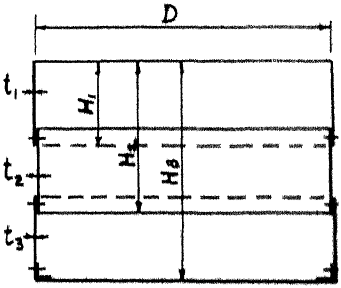


Fig. 8. Open Tank

The shell thickness is calculated for the levels corresponding to the different heads of liquid (H_1, H_2, H_3, in Fig. 8) above the lower edge of each course. The unit stresses used vary from 12,000 to 22,000 (oil) pounds per square inch, depending upon the character of the loading. The design of oil storage tanks is usually based on the specific gravity of water G = 1.

Spherical or elliptical bottoms are calculated like dished heads of pressure vessels, substituting $\frac{H}{2.309}$ for the working pressure p. In this case the bottom reinforcing ring is usually moved in so that it divides the surface of the bottom in equal parts: ring diameter D' = 0.709 D (diameter of tank).

Conical Tanks. - In a conical tank containing a liquid of specific weight γ the internal pressure at section mn distant h - y from the level of the liquid is,

$$q = (h - y)\gamma$$

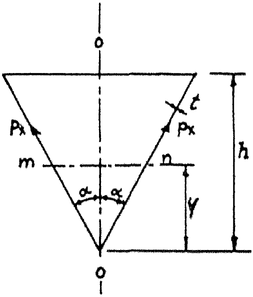


Fig. 9. Conical Tank

The hoop stress py obtains its maximum value for $y = \frac{h}{2}$, and is given by equation:

$$(py)_{max} = \frac{h^2 \cdot \gamma \cdot \tan\alpha}{4 \cdot t \cdot \cos\alpha} \qquad (2)$$

Stress p_x is calculated by equation:

$$p_x = \frac{y \cdot \tan\alpha \cdot \gamma}{2 \cdot t \cdot \cos\alpha}(h - \frac{2}{3}y) \qquad (3)$$

where t denotes the thickness of the shell.

Rectangular Tanks. - The maximum fiber stress in the plate of open rectangular tanks with a long side a and a small side b is

$$f = \frac{C \cdot H \cdot G \cdot}{2.309 \cdot t^2} \qquad (4)$$

where f = fiber stress, lb. per sq. in.
C = coefficient, depending on a/b ratio, see below,
a = long side of rectangle, in.
b = short side of rectangle, in.
Other notations as above.

By inserting the joint efficiency e and the maximum allowable working stress S, in lb. per sq. in., the thickness of the plate is obtained

$$t = b\sqrt{\frac{C_1 \cdot H \cdot G}{2.309 \cdot e \cdot S}} \qquad (5)$$

The deflection of the plate is

$$y = \frac{H \cdot b^4}{2.309 \cdot E \cdot t^3} \qquad (6)$$

where E = modulus of elasticity, lb. per sq. in.

The following coefficients are to be used in these equations

a/b	1.0	1.25	1.50	1.75	2.0	∞
C_1	.308	.399	.454	.490	.497	.500
C2	.0138	.0199	.0240	.0264	.0277	.0284

The maximum fiber stress in a flat stayed plate, having the staybolts placed at the corners of a square, is

$$f = \frac{.228\,H \cdot G \cdot a^2}{2.309 \cdot t^2} \qquad (7)$$

where a = side of the square, in.

The deflection of this plate is

$$y = \frac{.0284\,H \cdot G \cdot a^4}{2.309\,E \cdot t^3} \qquad (8)$$

Chapter X

DISHED HEADS AND SPHERES

Introduction. - Dished heads, in the closer meaning of the term, can be divided into heads, which are developed by the revolution of an ellipse, and ordinary dished heads whose meridian is built up of two circular arcs. The crown of dished heads is usually dished to a radius equal to the diameter of the vessel while the connecting arcs between crown and cylinder are of a short radius.

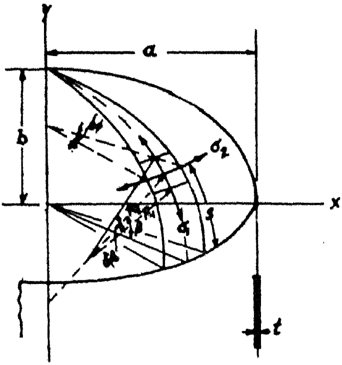


Fig. 10. Stresses in Dished Heads.

It has been customary, in the computation of dished heads, to compute merely the stress at the crown of the head, assuming it to be a sphere. This method fails to take into consideration the bending or "Discontinuity Stresses" which are produced by the mathematical discontinuity of the surface. If a pressure vessel consists of elements joined together which are different in shape, f.i. the connection of a dished head to a shell or the attachment of the spherical segment to the knuckle portion in a dished head, the stresses in adjoining portions will in general not be complimentary and there will be shear and bending stresses introduced.

In the case of the ordinary dished head the bending stresses due to the abrupt change in the radius of curvature of the head at the junction of the knuckle and the crown can become so great under the internal pressure that the head takes on a new shape in which the knuckle merges into the crown gradually, similar to the shape of an ellipsoid. In general a dished head tends to bulge outward at the crown under the influence of internal pressure and to pull in at the rim, whereby the radius of the crown decreases while the radius of the knuckle increases. The cylindrical shell, however, tends to expand in diameter. Since head and shell are joined together the result is that, along a meridional line, the shell is bent inwardly, and the rim of the head is bent outwardly. This results in considerable shear and bending stresses near the juncture, in addition to the membrane stresses. These stresses may occur not only in the knuckle of the head but also in the cylindrical shell. The smaller the knuckle radius is in comparison with the radius of the dome the higher the bending stresses will be. For this reason the A.S.M.E. Code specifies a knuckle radius of not less than 0.06 of the radius of the dish. Formulas for the computation of dished heads usually make allowance for the discontinuity stresses by applying a higher factor of safety or by using some other corrective factor.

Design of Dished Heads. - The stresses in dished heads subjected to internal pressure and the problem of their proper profile have been investigated quite extensively and a considerable amount of literature exists on this subject. Most of the early discussions have been written in German and the equations developed are presented in a rather inconvenient form. More recently a number of equations have been developed by Mr. W. M. Coates, which seem to be less cumbersome in their application.

Coates' Method. - Mr. W. M. Coates (ref. 1) developed a method of computing the maximum stresses in elliptical heads which consists in computing, first, the stresses that would exist in the shell and head of the vessel if they acted independently as true membranes. Thereupon the local discontinuity stresses are calculated, which are set up in the vicinity of the juncture of the head and shell due to the fact that these elements are joined together, and therefore, cannot assume their natural membrane displacements in this vicinity. These stresses are then combined algebraically to obtain the net stresses at any point.

The following formulas are given for the computation of the membrane stresses:

$$\frac{pR_2}{2t} \qquad (1)$$

$$\sigma_2 \quad \sigma_1\left[2 - \frac{R_2}{R_1}\right] \qquad (2)$$

in which σ_1 = meridional or radial stress, uniformly distributed over the thickness of the shell.
σ_2 = hoop stress.
p = internal pressure.
t = thickness of the shell.
R_1 = radius of curvature of the meridional section.
R_2 = radius of curvature of the section perpendicular to the meridian at the point in question (see Fig. 10).

Special Cases. -

(a) The Hemisphere: $R_1 = R_2 = a$ $\quad \sigma_1 = \frac{pa}{2t}$

$\sigma_2 = \sigma_1$

(b) The Cylinder: $R_1 \rightarrow \infty$, $R_2 = a$ $\quad \sigma_1 = \frac{pa}{2t}$

$\sigma_2 = 2\sigma_1$

(c) The Ellipsoid: $R_1 = kav^3$, $R_2 = kav$

$\sigma_1 = \frac{pka}{2t} \quad v$

$\sigma_2 = \sigma_1\left[2 - \frac{1}{v^2}\right]$

Where $k = \frac{a}{b}$, and $v = \dfrac{1}{\sqrt{1 + (k^2 - 1)\cos^2 \beta}}$

For the ellipsoid: a = radius of cylinder or semi-major axis of ellipse.
b = depth of head or semi-minor axis of ellipse.

Equation for the computation of the meridional displacement of any point P of a membrane under uniform inner pressure:

(a) For the hemisphere: $u = 0$ (3)

$$w = \frac{pa^2}{2Et}(1 - \mu) \tag{4}$$

(b) For the cylinder: $u = \frac{pas}{2Et}(1 - 2\mu)$ (5)

$$w = \frac{pa^2}{2Et}(2 - \mu) \tag{6}$$

(c) For the ellipsoid:

$$u = \frac{p \cdot a^2}{2 \cdot E \cdot t} \frac{\sqrt{k^2 - 1}}{2k} \cdot L \cdot \cos \beta \tag{7}$$

$$w = \frac{pa^2}{2Et}\left\{\frac{\sqrt{k^2 - 1}}{2k} L \sin \beta - k^2\left[(2 - \mu)v^2 - 1\right]\right. \tag{8}$$

in which u = component of the displacement along the tangent to the meridian.

w = component of the displacement along the normal to the surface.

E = modulus of elasticity.

μ = ratio of unit lateral strain to unit longitudinal strain (0.3 for steel).

S = distance measured along the meridian from the junction plane to the point in question.

β = angle formed by radius and large axis base line of the ellipsoid.

$$L = \left\{[2k^2 - (1 - 2\mu)] \log(k - \sqrt{k^2 - 1} \sin \beta)v \right.$$
$$\left. - [(1 - 2\mu)k\sqrt{k^2 - 1}]v^2 \sin \beta\right\}$$

Formulas for the computation of local discontinuity stresses:

(a) For the cylinder: $w = -\frac{pk^2a^2}{4Et} e^{-\lambda_o s} \cos \lambda_o s$ (9)

$$\sigma_{1e} \quad \frac{3pk^2}{4\lambda_o^2 t^2} e^{-\lambda_o s} \sin \lambda_o s \tag{10}$$

$$\sigma_2 = -\frac{pk^2a}{4t}\, e^{-\lambda_o s} \cos \lambda_o s \qquad (11)$$

$$\sigma_{2e} = \mu\sigma_{1e} \qquad (12)$$

$$\tau = -\frac{pk^2}{8\lambda_o t}\, e^{-\lambda_o s}\,(\cos \lambda_o s - \sin \lambda_o s) \qquad (13)$$

in which σ_{1e} = meridional stress due to bending on external surface.

σ_2 = hoop normal stress.

σ_{2e} = hoop stress due to bending on external surface.

τ = meridional shearing stress.

$$\lambda_o = \sqrt[4]{\frac{3(1-\mu^2)}{a^2t^2}} \qquad (13a)$$

The values of the functions with $\lambda_o s$ have been tabulated previously for the calculation of beams on elastic foundation (ref. 16 and 17) and are shown in Fig. 10a.

(b) For the Head: $$w = \frac{pk^2a^2}{4Et}\left(e^{-\int_0^s \lambda ds}\right)\cos\left(\int_0^s \lambda ds\right) \qquad (14)$$

$$\sigma_2 = \frac{pk^2a^2}{4tR_2}\left(e^{-\int_0^s \lambda ds}\right)\cos\left(\int_0^s \lambda ds\right) \qquad (15)$$

$$\tau = \frac{pk^2a^2\sqrt{2}}{8\lambda t R_2^2}\left(e^{-\int_0^s \lambda ds}\right)\cos\left(\frac{\pi}{4} + \int_0^s \lambda ds\right) \qquad (16)$$

$$\sigma_{1e} = \frac{-3pk^2a^2}{4\sqrt{3(1-\mu^2)}\,tR_2}\left(e^{-\int_0^s \lambda ds}\right)\sin\left(\int_0^s \lambda ds\right) \qquad (17)$$

$$\sigma_{2e} = \mu\sigma_{1e} \qquad (18)$$

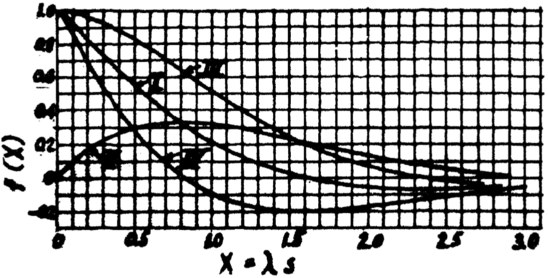
$$\lambda = \sqrt[4]{\frac{3(1-\mu^2)}{R_2^2\, t^2}} \qquad (18a)$$

In this term, R_2 is constant only for the hemisphere and for the cylinder, where $R_2 = a$ and $\lambda = \lambda_0$. In the case of dished heads, R_2 is variable and the equation has to be integrated numerically. Starting from the junction with the cylinder, a meridional arc is taken, so short that by regarding R_2 as constant for this arc no appreciable error is made. The fraction of the arc ϕs, and R_2 can be expressed in terms of β as follows:

$$\delta s = R_1 \delta\beta \quad \text{where } \delta\beta \text{ is in radians}$$

$$R_2 = kav = ka \,/\sqrt{1 + (k^2-1)\cos^2\beta}$$

Then the variable ratio $\delta s \,/\sqrt{R_2 (\text{mean})}$ is computed for each interval $\delta\beta$ and successive values are added cumulatively over the range investigated.



I. $f(x) = e^{-x} \cos x$
II. $f(x) = e^{-x} \sin x$
III. $f(x) = e^{-x} (\cos x + \sin x)$
IV. $f(x) = e^{-x} (\cos x - \sin x)$

Fig. 10a. Functions of (s).

The zero values of the functions with $e^{-\lambda_0 s}$ occur at an interval of π in the values of $\lambda_0 s$. Critical ordinates for any of these functions have a phase difference of $\pi/4$. For instance the function $e^{-\lambda_0 s} \sin \lambda_0 s$ has a maximum at $\pi/4$, and the function $e^{-\lambda_0 s} \cos \lambda_0 s$ has a zero value at $\pi/2$. The width of the zone between critical ordinates is

$$s = \pi/\lambda_o$$

For $\mu = 0.3$, this equation becomes $s = 2.44\sqrt{at}$.

Accordingly, the maximum value of the meridional bending (discontinuity) stress σ_{1e} in the cylinder occurs at a distance $\lambda_o s = \pi/4$, that is $s = 0.61\sqrt{at}$ from the junction with the head. The bending hoop stress σ_2 of the shell has its highest value at $s = 0$, or at the junction of head and cylinder, and becomes zero at the distance $1.22\sqrt{at}$. All of these bending stresses are so rapidly damped out that their values outside an edge zone of width $s = 2.44\sqrt{at}$ are of little importance. Welding seams should be located outside this zone.

The distance s is measured along the meridian from the junction of the shell and head to the point in question and is positive in the direction away from the junction. The displacement w normal to the surface is positive, when it is directed outward.

Discontinuity in Standard Dished Heads. - In standard dished heads, the meridian of which is made up by means of two radii, surface discontinuities exist at the junction of the crown and knuckle as well as at the junction of the knuckle and shell. Coates' method, which has been developed for elliptical heads, takes care of the discontinuity at the junction of knuckle and cylinder. In that case R_1 equals knuckle radius r and is constant. The hoop radius of curvature R_2 may be computed by the following equation

$$R_2 = r + (a - r)\sec\beta$$

For the calculation of the discontinuity stresses at the junction of knuckle and crown, C.O. Phys (ref. 14) suggests the following formulas:

(a) For the knuckle:

$$\sigma_1 = \frac{3\ p\ KA^2}{4\sqrt{3(1-\mu^2)}\,tR_2}\left(e^{-\int\lambda\,ds}\right)\sin\left(\int\lambda'\,ds\right) \qquad (18b)$$

$$\sigma_2 = \frac{pKA^2}{4rt}\left(e^{-\int\lambda\,ds}\right)\cos\left(\int\lambda\,ds\right) \qquad (18c)$$

(b) For the crown:

$$\sigma_1 = \frac{3pK}{4\lambda'^2t^2} (e^{-\lambda s}) \sin (\lambda' s) \qquad (18d)$$

$$\frac{-paK}{4t} (e^{-\lambda' s}) \cos (\lambda' s) \qquad (18e)$$

where A = radius of the crown, in.
a = radius of the cylindrical shell, in.
b = depth of the dished head, in.
K = (A/r) -1
$\lambda' = \sqrt{\frac{3(1-\mu^2)}{A^2t^2}}$
r = meridional radius of knuckle
Other designations as above

The highest discontinuity stress, which is set up in the head, is the bending meridional stress due to the crown-knuckle junction

$$\sigma_1 \max = 0.147 \, p \, A^2K/at$$

It occurs in the knuckle section. Experiments tend to support this theoretical consideration, because heads usually fracture within the knuckle.

Example: Given: A semi-elliptical head, radius of cylinder a = 50 in., depth of head b = 25 in., k = a/b = 2, wall thickness of head and cylinder t = 1.0 in., $\mu = 0.3$, E = 30000000 lb. per sq. in., internal pressure p = 100 lb. per sq. in.

Required: Membrane and discontinuity stresses and membrane displacements.

Solution: The membrane stresses and displacement in the cylinder are

$$\sigma_1 = 100 \times 50/2 \times 1.0 = 2500$$
$$\sigma_2 = 2 \times 2500 = 5000$$
$$w = 100 \times 50^2 \, (2-0.3)/2 \times 30000000 \times 1.0 = 0.00708$$

The membrane stresses and displacement in the head are then calculated by means of equations (1c), (2c) and (8) for several values of β. The results are tabulated in Table 14a.

Table 14a. - Membrane Stresses and Displacements of the Head.

(1)	β	0	5	10	15	20	25	30
(2)	$v = 1/\sqrt{1+(k^2-1)\cos^2\beta}$	0.500	0.501	0.506	0.513	0.523	0.538	0.555
(3)	$1/v^2$	4.00	3.99	3.91	3.80	3.65	3.46	3.25
(4)	$\sigma_1 = 5000\ v$	2500	2505	2530	2565	2615	2690	2775
(5)	$\sigma_2 = \sigma_1\ (2-1/v^2)$	-5000	-4980	-4840	-4600	-4310	-3930	-3470
(6)	$\frac{9.5}{\sqrt{3}} \log[(2+\sqrt{3}\ \sin\beta)v]$	0.000	0.406	0.834	1.257	1.663	2.12	2.55
(7)	$v^2 \sin\beta$	0.000	0.022	0.044	0.068	0.094	0.122	0.154
(8)	$L' = (6) - (7)$	0.000	0.384	0.790	1.189	1.57	2.00	2.40
(9)	$0.6\ L' \sin\beta$	0.000	0.020	0.082	0.185	0.32	0.51	0.72
(10)	$4(1.7\ v^2 - 1)$	-2.3	-2.29	-2.26	-2.21	-2.14	-2.03	-1.91
(11)	(9) + (10)	-2.3	-2.27	-2.18	-2.02	-1.82	-1.52	-1.19
(12)	$w \times 10^3 = 4.17 \times (11)$	-9.6	-9.47	-9.1	-8.43	-7.59	-6.34	-4.55

Table 14b. - Discontinuity Stresses of the Head.

(1)	β	0	5	10	15	20	25	30
(2)	v	0.500	0.501	0.506	0.513	0.523	0.538	0.555
(3)	$R_1 = kav^3$	12.5	12.61	12.97	13.50	14.31	15.57	17.07
(4)	R_1 (mean)	12.5	12.55	12.79	13.24	13.91	14.94	16.32
(5)	$\delta s = R_1 \text{ (mean)} \times \delta\beta$	0.000	1.095	1.118	1.158	1.216	1.306	1.423
(6)	$R_2 = kav$	50.0	50.1	50.6	51.3	52.3	53.8	55.5
(7)	R_2 (mean)	50.0	50.05	50.35	50.95	51.8	53.05	54.65
(8)	$\delta s \sqrt{R_2 \text{ (mean)}}$	0.000	0.155	0.157	0.162	0.169	0.179	0.193
(9)	$\Sigma \delta s \sqrt{R_2 \text{ (mean}}$	0.000	0.155	0.312	0.474	0.643	0.822	1.015
(10)	$x = 1.285 \times (9)$	0.000	0.199	0.401	0.609	0.826	1.057	1.304
(11)	$e^{-x} \cos x$	1.0	0.8	0.61	0.45	0.29	0.2	0.08
(12)	$e^{-x} \sin x$	0	0.17	0.26	0.31	0.32	0.31	0.27
(13)	$\sigma_2 = \frac{250000}{R_2(\text{mean})} \times (11)$	5000	3990	3030	2210	1400	943	366
(14)	$\sigma_{1e} = -\frac{454500}{R_2(\text{mean})} \times (12)$	0	-1545	-2350	-2760	-2810	-2660	-2250

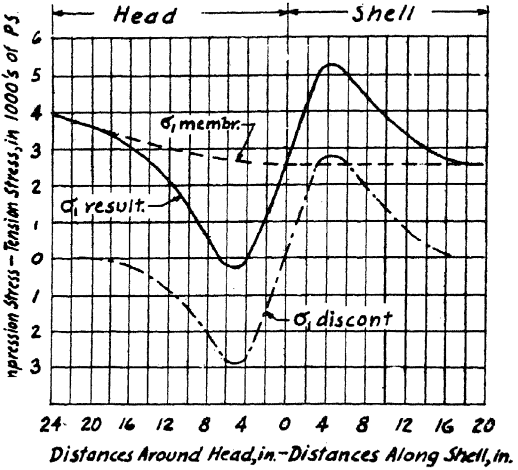


Fig. 10b. Meridional Stresses in 100" Dia. Elliptical Head.

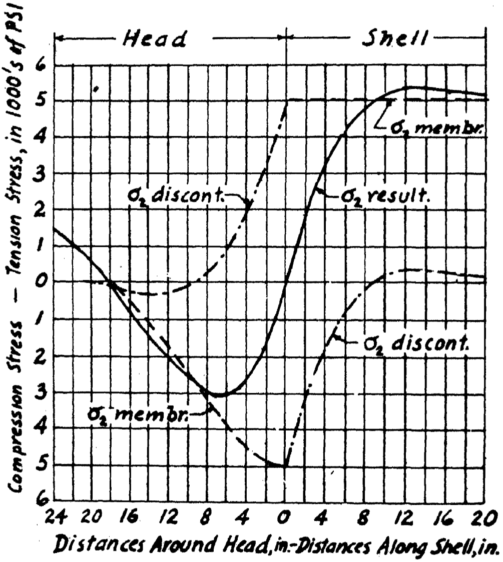


Fig. 10c. Hoop Stresses in 100" Dia. Elliptical Head.

The discontinuity stresses in the cylinder:

$\lambda_0 = 0.185$
$s = 2.44\sqrt{50x1.0} = 17.24$ (corresponding to $x=\pi$ in Fig. 10a)
$\sigma_{1e} = + 8760\ e^{-\lambda_0 s} \sin \lambda_0 s$
$\sigma_{1e}\ \text{max.} = + 2770$ (at $x = \pi/4$)
$\sigma_2 = - 5000\ e^{-\lambda_0 s} \cos \lambda_0 s$
$\sigma_2 = + 400$ (at $x = 0.75\ \pi$)

Values of the functions $(\lambda_0 s)$ are obtained from Fig. 10a.

The discontinuity stresses in the head are computed for several evenly increased values of β by means of equations (15) and (17). The necessary numerical integration is carried out in Table 14b. The resulting stresses are tabulated there too. A diagrammatical picture of the meridional and the hoop stresses is presented in Fig. 10b and Fig. 10c.

A.S.M.E. Code.

Shape. - (a) Standard dished heads: radius of the dish R not to be larger than the outside diameter of the head. The corner or knuckle radius r not to be less than 0.06 x diameter of the shell, or 3 x thickness of the head.

(b) Elliptical heads: depth of the dish = $\frac{1}{4}$ diameter of the head.

Standard dished heads usually have an even outside diameter, and elliptical heads an even inside diameter.

Thickness. According to the proposed 1942 revision of the A.S.M.E. code, the formulas specified by the A.P.I. - A.S.M.E. code, and presented further down in this text, are to be used in calculating head thicknesses. Under the previous rules, heads were computed as follows:

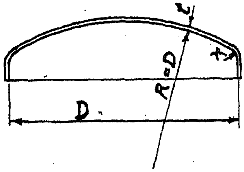


Fig. 11

Thickness:

(a) pressure on concave side.

$$t = \frac{8.33 \times P \times R}{2 \times TS} \qquad (19)$$

P = working pressure, lb. per sq. in.
R = radius of the dish.
TS = ultimate tensile strength.

If an opening, 6" dia. or larger, is in the head, increase t by 15%, or by at least 1/8". For hemispherical heads substitute 5 for 8.33 in eq. (19).

(b) pressure on convex side.

The maximum allowable working pressure on the convex side is 60% of the allowable working pressure on the concave side

$$T = \frac{8.33 \times P' \times R}{1.2 \times TS} \qquad (20)$$

P' = working pressure on convex side, lb. per sq. in.

This formula errs on the safe side for values of r/R larger than 0.1, but is considerably on the unsafe side for heads with values of r/R less than 0.06.

A.P.I.-A.S.M.E. Code. -

Shape. (a) Standard dished heads: mean radius of the dish not to be larger than the mean diameter of the shell; knuckle radius to be at least 6 per cent of the mean diameter of the shell, or three times the thickness of the head plate.

(b) Ellipsoidal: ratio of inside major axis a to inside minor axis b not exceeding 2.6.

Thickness. Pressure on concave side.
(1) For Ellipsoidal Heads

$$t = \frac{pD_mV}{2SE} + c \qquad (21)$$

(2) For Dished Heads

$$t = \frac{pR_eW}{2SE} + c \tag{22}$$

(3) For Conical Heads

$$t = \frac{pD_c}{2SE \cos \alpha} + c \tag{23}$$

(4) For Hemispherical Heads

$$t = \frac{pD_m}{4SE} + c \tag{24}$$

where t = thickness in inches.
p = maximum allowable working pressure in lb. per sq. in.
D mean diameter in inches of the head flange, before corrosion allowance is added.
S = maximum allowable working stress corresponding to the operating temperature, in lb. per sq. in.
c = allowance for corrosion in inches.
E = efficiency of any joint in the head.
V = value obtained from Fig. 12,

$V = 1.55$ for $a/b = 2.6$
$V = 1.0$ for $a/b = 2.0$

W = value obtained from Fig. 13,

$W = 1.77$ for $r/R = 0.06$

R_c = the radius in inches of the crown measured to the center line of the thickness of the crown plate.
α = one-half the included angle of the cone at the center line of the head.
D_c = mean diameter in inches of the cone at the point under consideration, before corrosion allowance is added.

Figs. 12 and 13 have been redrawn from the A.P.I.-A.S.M.E. Code.

Pressure on convex side. Divide thickness, obtained by the appropriate formula, by 0.6.

Flat Heads. See under Cover Plates.

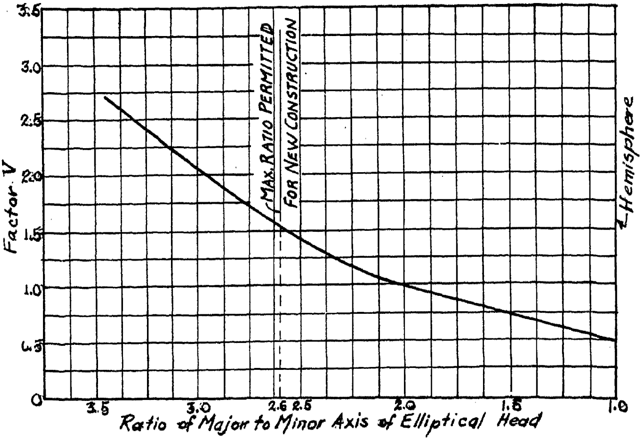


Fig. 12. Values of Factor V.

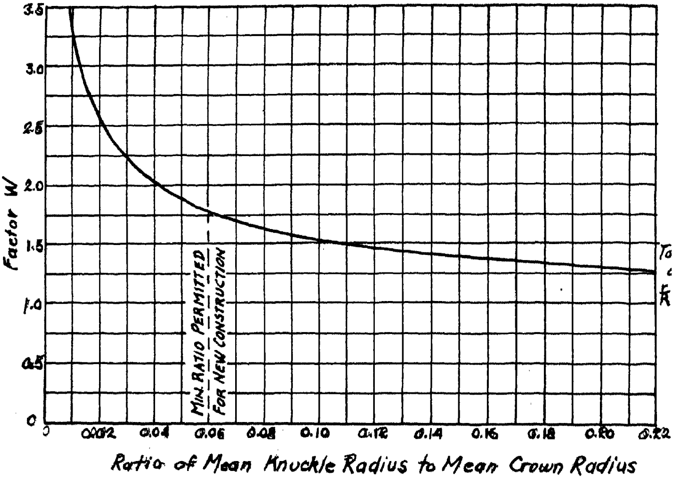


Fig. 13. Values of Factor W.

Hollow Spheres. -

1. Internal Pressure.

(a) Thick spheres:

$$r_2 = r_1 \sqrt[3]{\frac{2\mu S + 2(\mu - 2)p_1}{2\mu S - (\mu + 1)p_1}} \tag{25}$$

For $\mu = 0.3$

$$r_2 = r_1 \sqrt[3]{\frac{S + 0.4\, p_1}{S - 0.65 p_1}} \tag{25a}$$

where r_1 = inside radius, in.
r_2 = outside radius, in.
S = allowable working stress, lbs. per sq. in.
p_1 = internal pressure, lbs. per sq. in.

(b) Thin spheres:

$$t = \frac{d_1 p_1}{4S} \tag{26}$$

where d_1 = inside diameter, in.

2. External Pressure.

(a) Thick spheres:

$$r_2 = r_1 \sqrt[3]{\frac{2mS}{2mS - 3(m - 1)p_2}} \tag{27}$$

For $m = \frac{10}{3} = \frac{1}{\mu}$

$$r_2 = r_1 \sqrt[3]{\frac{S}{S - 1.05\, p_2}} \tag{27a}$$

where p_2 = external pressure, lbs. per sq. in.

(b) Thin spheres:

$$t = \frac{d_2 p_2}{4S} \tag{28}$$

where d_2 = outside diameter, in.

Note: If the sphere is produced by riveting or welding, the efficiency of the joint has to be taken into consideration.

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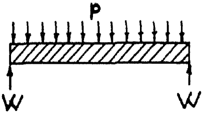
Chapter XI

FLAT HEADS AND COVER PLATES

Introduction. - It is very often necessary to close openings on pressure vessels by bolted on cover plates or heads. In general, a dished head or a flat plate bumped in the center provide greater strength and require less thickness. However, this type of construction is more expensive and, particularly for small openings, it is cheaper to use a simple flat plate made thick enough to resist the bending stresses.

Uniform Pressure p, lbs. per sq. in.



W, lbs. per lin. in.

Fig. 14. Circular Plate With a Uniform Pressure, Simply Supported

Stresses in Flat Plates. - For circular flat heads two conditions of support can be distinguished. In the first case, which rarely exists, the plate is simply supported without any bending moment at the edges (Fig. 14). In the second case the edges of the plate are fixed against rotation (as might result from being clamped between two rigid flanges). Thereby a bending moment is produced at the edge which tends to maintain the slope zero (Fig. 15).

Uniform load p, lbs. per sq. in.

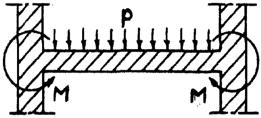


Fig. 15. Circular Plate With Uniform Pressure, Fixed at the Edge

In bolted cover plates with a tongue and groove joint the loading consists of the uniform load across the plate inside the gasket surface, due to the fluid pressure, and a load caused by the gasket pressure which is applied in a narrow ring along the face of the gasket. These two pressures are resisted by the tension of the bolts which can be assumed to be uniformly distributed along the bolt circle (Fig. 16).

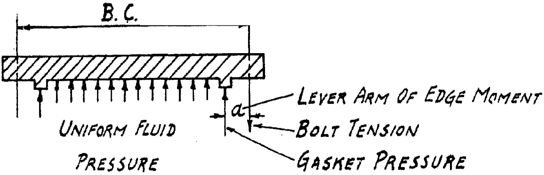


Fig. 16. Flat Circular Cover for Flanged Openings

Design of Flat Plates Under Uniform Load. - The following formulas are available for the computation of the thickness of flat plates under uniform pressure:

Circular Flat Plates. -

(1) Grashof's formula (Kent's Handbook)

$$t = r\sqrt{\frac{5}{6}\,\frac{p}{s}} \tag{1}$$

This formula is based on the Maximum Strain Theory.

(2) C. Von Bach's formula (Mark's Handbook)

$$t = r \tag{2}$$

$C = 0.75$ for mild steel
$= 1.2$ for cast iron

(3) Timoshenko and Lessels' formula (Applied Elasticity," pp. 278)

$$t = r\sqrt{\frac{3}{8}\,\frac{p}{s}\,(3 + \mu)} \tag{3}$$

$$t = r\sqrt{1.25\frac{p}{S}}\text{, for } \mu = 1/3 \qquad (3a)$$

where t = thickness, inches
r = radius, inches
p = uniform pressure, lbs. per sq. in.
S = stress, lbs. per sq. in.
μ = Poisson's ratio, (= 1/3 for steel)

These formulas are for flat plates simply supported at the edge. Timoshenko's formula gives a thickness 29% greater than that given by Bach's formula for a mild steel plate, and 22% greater than that given by Grashof's formula.

For fixed edges and $\mu = 1/3$, Timoshenko's formula is written:

$$t = r\sqrt{\frac{3}{4}\,\frac{p}{S}} \qquad (4)$$

Elliptical Flat Plates. -

$$t = \sqrt{\frac{Ka^2b^2p}{(a^2 + b^2)\ S}} \qquad (5)$$

where $2a$ = major axis
$2b$ = minor axis
K = factor depending on type of support

Simply supported plates:

K = 2 for steel
= 2.25 for cast iron

For plates fixed around the circumference

K = 1.33 for steel
= 1.5 for cast iron.

Square Flat Plates. - The maximum fiber stress with supported edges is

$$S = \frac{0.2208\ p \cdot a^2 \cdot (m + 1)}{m \cdot t^2} \qquad (6)$$

where a = side of the square, in.
$m = 1/\mu$, reciprocal of Poisson's ratio

For fixed edges

$$S = 0.308\ p \cdot a^2/t^2 \tag{7}$$

Rectangular Flat Plates. - If the edges are supported, the maximum fiber stress, in lb. per sq. in., is

$$S = \frac{0.75\ p \cdot b^2}{(1 + 1.61K^3) \cdot t^2} \tag{8}$$

where a = long side of rectangle, in.
b = short side of rectangle, in.
$K = b/a$

For a rectangular plate with fixed edges the maximum fiber stress is

$$S \quad \frac{0.5\ p \qquad}{(1 + 0.623\ K^6)\ t^2} \tag{9}$$

A.S.M.E. Code. - The A.S.M.E. Code for Unfired Pressure Vessels gives the following methods for the determination of the thickness:

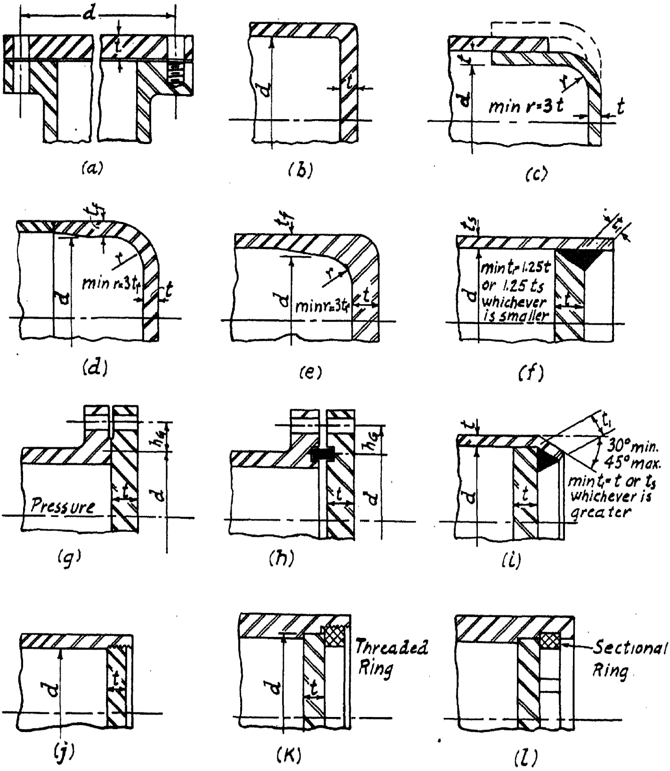
(a) Flat Cover Plates:
  (1) For diameters of 14 in. or less, use standard blind flanges (A.S.A. B 16c)
  (2) For diameters greater than 14 in., the following formula is used:

$$t = d\sqrt{\frac{C\ p}{S}} \tag{10}$$

where t = thickness in inches,
d = diameter in inches,
p = maximum allowable working pressure in lb. per sq. in.,
S = maximum allowable working stress, lbs. per sq. in.
C = constant depending on type of joint.

Values of C:

0.162	Cover plates, plain, rigidly riveted or bolted to shell, Fig. 17a.
0.162	Flat heads, integral with shell, maximum diameter d = 24 in., mimimum ratio t/d = 0.05; Fig. 17b.



(a) (b) (c) (d) (e) (f) (g) (h) (i) (j) (K) (l)

Fig. 17. Acceptable Types of Flat Heads and Covers

0.25 Flat heads, forged integral with shell; Fig. 17e.
Flat heads, butt-welded to shell; Fig. 17d.

0.30 Flat flanged heads, riveted or lap-welded to shell; Fig. 17c.
Flat plates inserted into the end of a vessel and held in place by some suitable positive mechanical locking arrangement. Fig. 17k and 17ℓ.

$0.30 + \frac{1.40 \times W \times h_G}{H \times d}$ cover plates with raised face or ring gasket, bolted to shell, Fig. 17g and 17h.

where W = total bolt load in lb.,
h_G = radial distance from the bolt circle diameter to the diameter d in inches,
H = total hydrostatic end force on area bounded by the outside diameter of the gasket or contact surface in lb.

0.50 Plates inserted into vessels and welded thereto, end of the vessel crimped over, maximum diameter d = 18 in., Fig. 17i.

0.75 Plates screwed into the end of a vessel, maximum diameter d = 12 in., Fig. 17j.

The figures have been redrawn from Fig. U-2 of the A.S.M.E. Unfired Pressure Vessel Code.

(b) Dished Cover Plates, containing a spherical segment:

$$S = \frac{3}{\pi(t - c)^2}\left(\frac{0.18\ W\ (r^2 - a^2)}{d^2} + 1.48\ W \log\frac{r}{a}\right) + \frac{R\ p}{2(t - c)} \qquad (11)$$

where S = maximum allowable working stress in lb. per sq. in.,
t = thickness in inches,
W = bolt load, which may be assumed as twice the maximum allowable working pressure (lb. per sq. in.) multiplied by the area (sq. in.) upon which the pressure is exerted,

a = distance from center line of cover plate to center line of gasket in inches,
r = radius of bolt circle in inches,
d = outside radius of cover plate in inches,
R = radius of spherical segment in inches,
p = maximum allowable working pressure in lb. per sq. in.,
c = corrosion allowance in inches.

Tube Sheets. - The tube sheets of heat exchangers with U-tubes or floating heads are subjected to a uniform pressure equal to the difference of the pressure on the shell and the tube side. In heat exchangers with fixed tube sheets, the forces have also to be considered, which are produced through the relative expansion or contraction of the tubes due to temperature changes and pressure. External pressure acting on a cylinder with open ends tends to increase its length, while internal pressure shortens it. Thus, condenser tubes, which have a higher external pressure, are elongated. A difference in temperature and coefficient of thermal expansion causes an uneven elongation of the shell and tubes. Depending on the direction of the forces, the expansion of the tubes may either aggravate or relieve the deflection of the tube sheet due to the working pressure.

The stresses are computed as follows:

I. U-Tubes and Floating Tube Sheets
   (a) Round Tube Sheet.
      1. Unstayed, ring gasket
$$s = 1.32\ r^2p/t^2 \qquad (12)$$
      2. Unstayed, full-faced gasket
$$s = 1.04\ r^2p/t^2 \qquad (13)$$
      3. Stayed, ring gasket
$$s = 1.32\ r^2p'/t^2 \qquad (14)$$
      4. Stayed, full-faced gasket
$$s = 1.04\ r^2p'/t^2 \qquad (15)$$
   (b) Rectangular Tube Sheet.
      1. Unstayed, ring gasket
$$s = k_1\ b^2p/t^2 \qquad (16)$$
      2. Unstayed, full-faced gasket
$$s = k_2\ b^2p/t^2 \qquad (17)$$

3. Stayed, ring gasket

$$s = k_1 b^2 p'/t^2 \tag{18}$$

4. Stayed, full-faced gasket

$$s = k_2 b^2 p'/t^2 \tag{19}$$

II. Fixed Tube Sheets'

(a) Round Tube Sheet

$$s = 2\ r^2 p A_1/At^2 \tag{20}$$

(b) Rectangular Tube Sheet

$$s = 2b^2 p A_1/At^2 \tag{21}$$

Equations (20) and (21) apply only to tubes, which are secured to the tube sheet by means of ferrules, but may also be used as a near approach in cases where the tubes are expanded or welded into the tube sheet.

Nomenclature

A = total area of tube sheet within center line of gasket, sq. in.
A_1= net area of tube sheet subjected to pressure (area A less area of all tube holes), sq. in.
a = mean length of gasket (rectangular tube sheets), in.
b = mean breadth of gasket (rectangular tube sheets), in.
k_1= coefficient for ring gasket (rectangular tube sheets)
k_2= coefficient for full-faced gasket (rectangular tube sheets)
p = maximum working pressure, lb. per sq. in.
$p' = (n/n+m)\ p$ = net working pressure for stayed tube sheets, lb. per sq. in.
total net area of flange bolts, sq. in.
total net area of stays, sq. in.
mean radius of ring gasket
allowable working stress in tube sheets, lb. per sq. in.
minimum thickness of tube sheet, in.

Table of "K" Values

b/a	1.0	0.9	0.8	0.7	0.6	0.5	0.4	0.3	0.2	0.1
k_1	0.346	0.418	0.494	0.569	0.638	0.700	0.748	0.784	0.804	0.810
k_2	0.359	0.434	0.504	0.557	0.597	0.627	0.646	0.659	0.665	0.669

Ring gaskets extend only to the inner line of the bolt holes.

The thickness of tube sheets is also governed by practical requirements. It should be in proper proportion to the diameter of the tubes and provide sufficient depth and strength for expanding and rolling of the tubes. The thickness of the tube sheet is usually not smaller than the outside diameter of the tubes: For copper base alloys the thickness is made equal to the outside diameter of the tube plus 1/8 in.

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Chapter XII

THE REINFORCEMENT OF OPENINGS

Introduction. - It is obvious that the removal of metal from the walls of pressure vessels for the purpose of providing connections or openings has a certain weakening effect. This weakening effect is intensified by the fact, established by experiments, that high stress concentrations take place in the metal around openings which may be 3 to 4 times as large as the stresses in solid shells, (Ref. 1). The stress concentration varies with the size and shape of the hole and with the type of the connection and is usually expressed by the stress concentration factor f, which is the ratio of the high stress s_h, to the stress in the normal or plain portion of the shell s_a, or

$$f = \frac{s_h}{s_a}$$

Stresses Around Openings. - The mathematical analysis of the stresses around openings has been attempted by several authors but it is very complicated and difficult to apply to practical conditions. Inglis (Ref. 2) derived the following formula for the maximum circumferential tension at the edge of an elliptical opening assuming that a uniform stress acted at the edge of the flat plate in one direction only, and that one axis of the ellipse be parallel to the direction of stress.

$$s_h \quad s_a\left(1 + \frac{2a}{b}\right) \tag{1}$$

where a = length of semi-axis perpendicular to the direction of stress on edge of plate,
b = length of semi-axis parallel to the direction of stress on edge of plate.

This formula gives the following stress concentration factors:

11" x 15" manhole: $f = 3.73$
10" x 16" $f = 4.20$
circular opening: $f = 3.00$

For a small elliptical opening in the center of a pressure vessel head the maximum stress concentration is equal to (2a/b) times the uniform stress. Where a nozzle or pipe is connected to a pressure vessel, the wall of the nozzle neck or pipe has a certain reinforcing effect on the metal around the opening. The stresses are highest at the edge of the opening and decrease very rapidly as the distance from the opening increases.

Design of Reinforcement. - In pressure vessel design account has been taken of the higher stresses by increasing the wall thickness if openings have to be provided in the shell. A more economical method is to attach a reinforcing pad by welding, riveting or forging to the vessel at the point where it is weakened, the cross-sectional area of which is made approximately equal to that of the metal removed.

The total cross-sectional area of the reinforcement within the limits of the "area of reinforcement" shall be at least:

(a) for riveted construction

$$A_1 + A_2 = [(d + 2c) + Nd_1](t - c)E \qquad (2)$$

(b) for welded and forged construction

$$A = (d + 2c)(t - c)E \qquad (3)$$

where A = area which is to be replaced by reinforcement, in sq. in.,
A_1 = area of the main opening in the vessel wall to be replaced by reinforcement, in sq. in.,
A_2 = area of rivet holes in the vessel wall to be replaced by reinforcement, in sq. in.,
d = diameter of the main opening in the vessel wall in inches,

d_1 = diameter of rivet holes in inches,
c = corrosion allowance in inches,
t = thickness of shell or plate in inches,
N = number of rivets intersected by the area of reinforcement,
E = longitudinal joint efficiency.

See Fig. 18 and 19 which have been redrawn from Fig. 16 and 27 of the A.P.I.- A.S.M.E. Code.

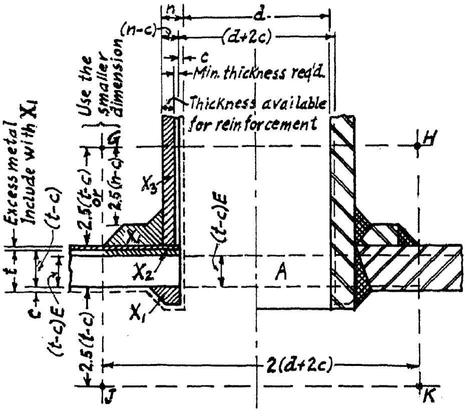

Fig. 18. Reinforcement of Single Opening (Welded)

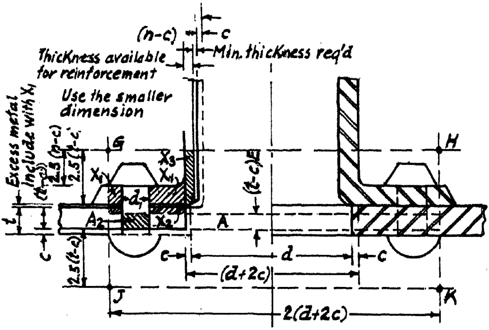
(n-c)
c
Thickness available for reinforcement
Min. thickness req'd
Use the smaller dimension
Excess metal Include with X1 (t-c)
2.5(n-c)
2.5(t-c)
G
H
X3
X1
X4
(t-c)E
d1
A2
X2
A
t
c
2.5(t-c)
c
d
c
(d+2c)
J
K
2(d+2c)

Fig. 19. Reinforcement of Single Opening (Riveted)

The area of reinforcement is represented by the rectangle GHKJ. Areas X_1, X_2, etc. are available for reinforcement.

The reinforcing pad should be strong enough to resist bending and at least from 3/4" to 1" thick. Various types of pads are employed. The ideal design of the reinforcement would be one similar to Fig. 20 where, as the edge of the opening is approached, the total thickness of the vessel increases in accordance with a hyperbolic function or some other function of the second degree, the reinforcement preferably being equally divided on both sides of the plate. However, this requires a construction which is expensive and impractical.

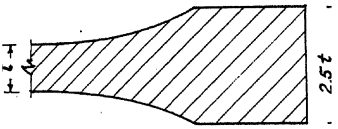

Fig. 20

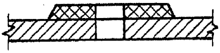

Fig. 21. Reinforcement of Small Openings by Pad of Welding Material

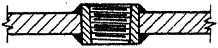

Fig. 22. Reinforcement of Small Openings

In general practice, small openings are reinforced by building up a pad completely of welding material, (Fig. 21) or by welding in a pipe coupling or spud, (Fig. 22). For large openings a plate, at least 3/4" to 1" thick, is cut and formed to fit, and welded in place as shown, in Fig. 23. The nozzle neck, which is usually made of approximately the same thickness as the vessel wall, must be welded to the shell and to the reinforcing ring which has to be welded to the shell also. In the case of heavy vessels frequently double reinforcing plates are used, as shown in Fig. 24. A Taylor welded

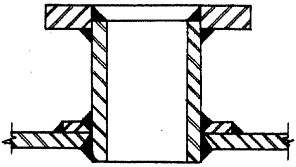

Fig. 23. Reinforcement of Opening by Welded Plate

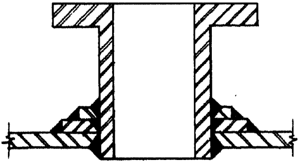

Fig. 24. Reinforcement of Openings with Multiple Plates

nozzle with extended neck is shown in Fig. 25 while Fig. 26 illustrates a sweep type nozzle.

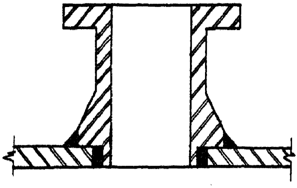

Fig. 25. Welding Type Nozzles with Extended Neck

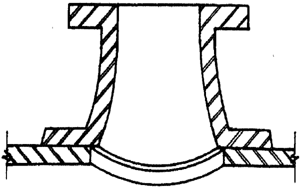

Fig. 26. Sweep Type Nozzles

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Chapter XIII

FLANGE DESIGN

Bolt Load. - In flange design the bolt load is usually taken as a starting point and then the flange is proportioned to stand the stresses set up by the bolt load. The total bolt load is equal to the total cross-sectional area of the bolts, measured at the smallest diameter in sq. in., multiplied by the maximum allowable unit working stress in the bolts:

$$W = A_B \times S_B \tag{1}$$

where W = total bolt load, lb.
A_B = total cross-sectional area of bolts, sq. in.
S_B = maximum allowable bolt stress, lb. per sq. in.

The bolt load has to be strong enough to resist the hydrostatic end force and to maintain a certain pressure on the gasket surface in order to produce the sealing effect.

Hydrostatic End Force. - The Hydrostatic End Force is equal to the working pressure times the area enclosed by the mean diameter of the contact surface.*

$$H = P \times A_p \tag{2}$$

where H = total hydrostatic end force, lb.
P = working pressure, lb. per sq. in.
A_p = area enclosed by the mean diameter of the contact surface, sq. in.

When the bolts of the flange are tightened initial stresses are set up before any internal load is applied. When the internal pressure is applied the load on the gasket will be decreased. The load on the bolts will be increased and should be the sum of the working pressure and the decreased gasket pressure.

***Note Preface to third edition.**

Gasket Load. - The Gasket or Contact Load is equal to the product of the area of the gasket surface and the unit gasket pressure. The unit gasket pressure is a multiple of the internal working pressure, and depends upon the material and shape of the gasket, type of flange joint, the nature of the confined fluid, and similar considerations. It must be greater than the fluid pressure in order to maintain tightness.

Contact-Pressure Ratio. - The ratio of the gasket pressure to the working pressure is known as the contact-pressure ratio.

The following values are used by many designers:

Type of Joint	Contact-Pressure Ratio
Small tongue and groove	12
Wide male and female	6
Raised face	3

The question how much the gasket pressure should exceed the working pressure has not been sufficiently investigated yet. Another difficulty exists in the determination of the actual gasket pressure set up in a bolted connection. It is not unusual or especially difficult to apply, with wrenches of the usual length, sufficient force to stress bolts to the elastic limit. For example, in the case of an 8", Series 15, flange with large T. & G. joint, the foot pounds required to set up the assumed gasket load is 88 which means that one man using a 1 ft. wrench can easily exert this force. Wherever practical it, therefore, should be attempted to measure or control the effort applied to tightening up bolts.

On account of the many indeterminate factors involved, the gasket load is usually expressed as follows:

$$H_G = W - H \qquad (3)$$

where H_G = total gasket or contact load, lb.
W and H as above.

Bolts. - An approximate assumption of the diameter of the bolt circle can usually be made from the inside diameter of the flange which has to be known. The number and size of the bolts then have to meet the following requirements:

1. The total allowable bolt stress has to be equal to the sum of the load due to the fluid pressure and the load due to the desired gasket pressure.
2. The bolts have to be spaced so that ample nut clearance is provided, for use of socket or open-end wrenches. A distance of the bolts, measured center to center, equal to four times the bolt diameter is usually sufficient. Bolts up to 1 inch diameter are spaced at approximately 3 inch centers. The number of bolts is usually a multiple of 4.

Allowable Material Stresses. - The A.P.I.-A.S.M.E. Code specifies a design working stress of 11,000 lbs. per sq. in. for carbon steel bolts (temperatures not over 450°F), and for alloy-steel bolts the stresses allowed for plates having an ultimate tensile strength of 60,000 lbs. per sq. in. Based on a factor of safety of 4, the working design stress is 15,000 lbs. per sq. in. at normal temperatures. The allowable stress for alloy-bolts is considered too low by many designers and stresses of 20,000 to 25,000 lbs. per sq. in. are frequently used.

Flanges are made of materials which have an ultimate tensile strength varying from 45,000 to 70,000 lbs. per sq. in. The most frequently specified material is a carbon-steel having an ultimate tensile strength of 60,000 lbs. per sq. in.

Flange-Stress Calculation. - Flanges with Gasket Extending Full Width. - No bending moments occur under atmospheric pressure. Under working pressure the flange can be compared to a simple beam, with a concentrated load at the center (see Fig. 27), where M max = PL/4. If we consider the flange as made up of a number of sectors equal to the number of bolts, the width of the beam can be assumed as equal to the distance between two adjoining bolt holes or b - d. Further, for approximate purposes a is considered equal to L/2. This gives the equation:

$$H'\frac{(B - A)}{2} = \frac{(b - d)t^2S}{6} \qquad (4)$$

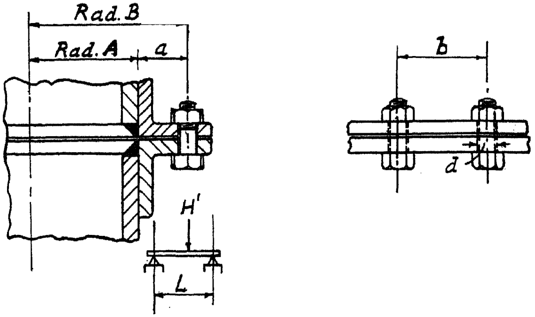


Fig. 27

or

$$t = 1.73\sqrt{\frac{H'(B - A)}{S(b - d)}} \qquad (5)$$

where A = inside radius of vessel, in.
B = radius of bolt circle, in.
a = lever arm of bolt load, in. = $B - A$.
b = bolt pitch, in.
d = diameter of bolt hole, in.
H' = hydrostatic end force per bolt, lb.
S = unit design stress of flange, lb. per sq. in.
t = thickness of flange, in.

The value to be used for the hydrostatic end force should be about 1.25 times the actual hydrostatic end force in order to make allowance for the gasket pressure. H' can also be computed as the sum of the actual hydrostatic pressure (area of mean gasket dia. X working pressure) plus the gasket pressure (gasket area X working pressure X 2 or 3), divided by the number of bolts.

Flange with Contact Surface within Bolt Circle.- In flanges which have the gasket or contacting flange surfaces entirely within the bolt circle, such as loose ring flanges, raised face flanges, tongue and groove joints, etc., bending stresses are set up through an external moment which is equal to the total bolt load multiplied by its lever arm.

The lever arm of loose flanges is represented by the radial distance from the bolt-circle diameter to the line of pressure of the opposing force, which is considered as located in the middle of the gasket surface.

Flanges which are integral with the vessel or nozzle neck (welded, riveted, screwed, or forged) involve a compound lever arm, which is calculated by the following formula:

$$a = \frac{H_D h_D + H_T h_T + H_G h_G}{W} \tag{6}$$

where H_D end force due to hydrostatic pressure on area bounded by inside diameter B, lb.
h_D = lever arm of force H_D, equal to radial distance from bolt-circle diameter to the line of pressure of force H_T, in.
H = total hydrostatic end force as defined above, lb.
H_T = end force due to hydrostatic pressure on flange ring area bounded by mean diameter of contact surface and inside diameter, equal to $H - H_D$, lb.
h_T = lever arm of force H_T, equal to radial distance from bolt-circle diameter to the line of pressure of force H_T, in.
H_G = total gasket or contact load as defined above, lb.
h_G = lever arm of force H_G, equal to radial distance from bolt-circle diameter to the line of pressure of force H_G, in.
W = total bolt load as defined above, lb.

The location of the different forces and their respective lever arms are shown for several types of flanges in Fig. 28, which has been redrawn from Fig. UA-2 of the A.S.M.E. Code. Several types of flanges are shown in Fig. 29, and types of flange faces in Fig. 30.

For the calculation of flange stresses several methods have been suggested which can be classified into two general groups. The first group comprises formulas which are based on the "Maximum Stress" theory and are of a rather approximate character. Representative of this group is the "Rankine Method." The second group is based on the "Maximum Strain" theory. This method, which is more accurate, but also more cumbersome, was

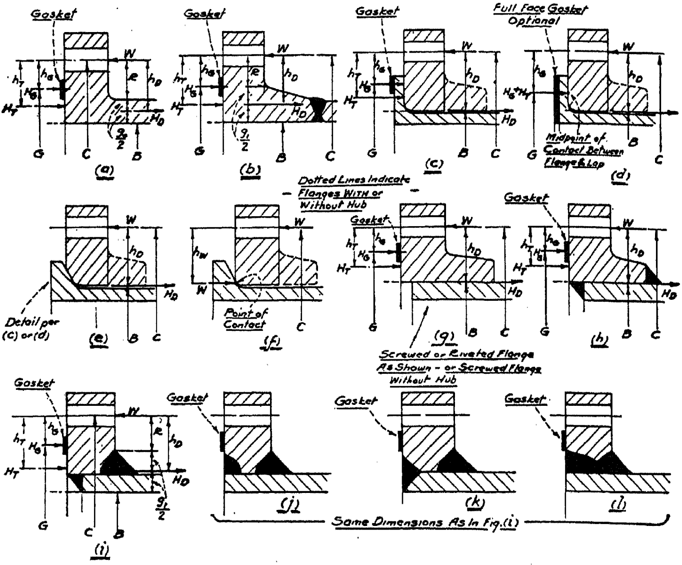
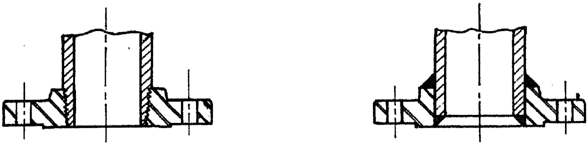

Gasket
Full Face Gasket Optional
Midpoint of Contact Between Flange & Lap
Dotted Lines Indicate Flanges WITH or Without Hub
Detail per (c) or (d)
Point of Contact
Screwed or Riveted Flange As Shown - or Screwed Flange Without Hub
Same Dimensions As In Fig.(i)
W
h_T
h_G
h_D
H_G
H_T
H_D
H_G+H_T
h_W
R
$\frac{g_1}{2}$
G
C
B
(a)
(b)
(c)
(d)
(e)
(f)
(g)
(h)
(i)
(j)
(k)
(l)

Fig. 28. Types of Bolted Flanged Connections

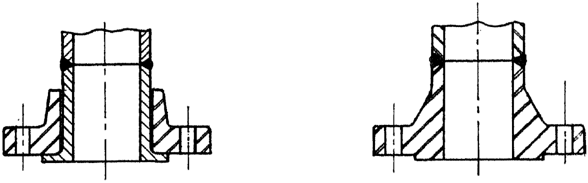
adopted by Waters and Taylor in their investigation (ref. 2). An improved version of this method, which is also applicable to flanges with tapered hubs, was later presented by a subgroup of the Joint A.P.I.-A.S.M.E. Committee for Bolted Flanged Connections (ref. 8).

Rankine Method. - A good account of this method is given in the Piping Handbook by J. H. Walker and S. Crocker. The following two assumptions are made: (1) the bolt forces, which are usually assumed as acting



Screwed Flange

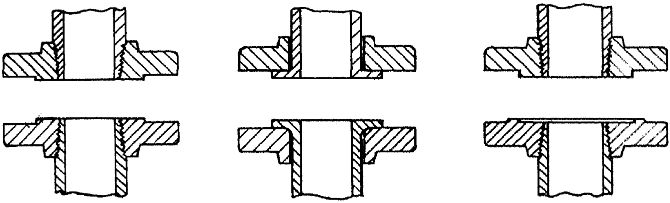
Slip-on Welding Flange



Van Stone Flange

Welding Neck Flange

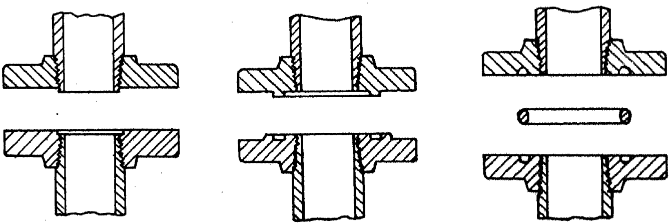
Fig. 29. Types of Forged Steel Flanges.



Raised Face

Van Stone

Large Male and Female



Small Male and Female

Tongue and Groove

Ring Joint

Fig. 30. Typical Flange Facings.
(American Standard; except ring joint.)

along the bolt circle, shift toward the inner edge of the bolt holes when load is applied. (2) The gasket reaction tends to shift to the outside of the raised surface as the flange is deflected through the bending moment.

(a) Flanges which fail in a circular section.
Due to the moment of the deforming couple the flange tends to fail at B (Fig. 31). The stresses in the Circular Section B - B' may be determined by the following formula:

$$S = \frac{6Wa}{bt^2} - \frac{6A_BS_Ba}{bt^2} \tag{7}$$

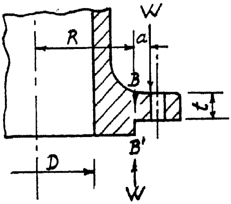


Fig. 31

where S = flange stress in section B - B', lb. per sq. in.
W = bolt force, lbs. = $A_B \times S_B$.
A_B = total bolt root area, sq. in.
S_B = bolt stress, lb. per sq. in.
a = moment arm from inner edge of bolt hole to section B - B', in.
b = breadth of beam = $2\pi R$.
t = depth of beam = flange thickness.

(b) Flanges which fail in a diagonal section.
Flanges and fittings with a small male and female joint tend to fail in the section C - C' (Fig. 32). The external moment composed of bolt forces and gasket reaction is in this case the same as above. The internal moment of the rectangular section of depth h and width b can be expressed:

$$M = \frac{pbh^2}{6}$$

$$M \quad \begin{matrix} pbd^2 \\ 6\cos^2\alpha \end{matrix} \qquad \text{for } h = \begin{matrix} \\ \cos\alpha \end{matrix}$$

By equating the external and internal moments, expressed per unit length of circumference, the following equation for the stresses in the diagonal ring section C - C' is obtained:

$$S = \frac{0.96 \cdot \cos^2 \alpha \cdot A_B \cdot S_B \cdot a}{R \quad {}^{2}} \qquad (8)$$

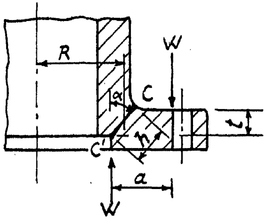


Fig. 32

where S = flange stress in section C - C', lb. per sq. in.
W = total bolt load = A_B x S_B, lb.
A_B = total bolt area, sq. in.
S_B = bolt stress, lb. per sq. in.
a = moment arm of bolt load W, in.
R = radius of mean circle of section C - C'.
α = angle between center line of fitting and slope of section C - C' (obtained by laying out the flange to scale)
h = depth of section C - C', in.
t = minimum flange thickness, in.

Method of the Joint A.P.I.-A.S.M.E. Committee. - For the investigation, the tapered-hub flange was subdivided into three portions: ring, hub, and shell. Each part was then studied separately, the boundary conditions being identical after assembly. While in prior investigations only the tangential stresses at the inner diameter of the flange had been checked, here the following types of loading were considered:

(a) the internal hydrostatic pressure acting in radial and axial direction,
(b) the external moment acting on the ring.

The division of the flange and the location of the forces is shown in Fig. 33. It was further assumed that hub and shell would act as membranes subject to tension and bending. The stresses were then determined by the strain-energy method, treating the hub as a beam with varying sections on an elastic foundation, and the boundary conditions evaluated.

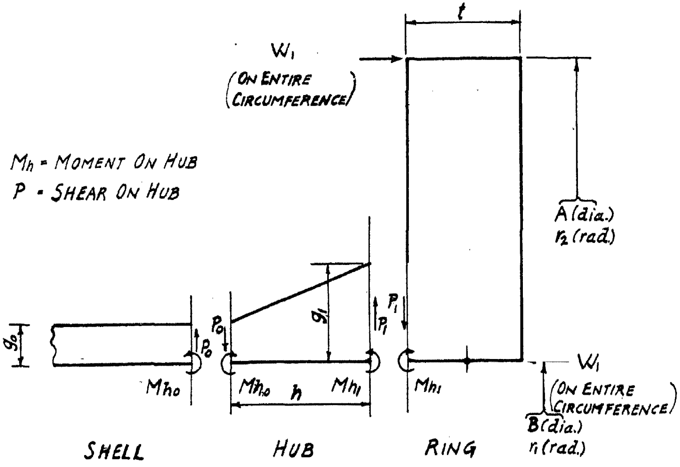


Fig. 33. Analysis of Forces and Moments in a Tapered-Hub Flange.

The results of the analysis indicated that in a flange with a straight hub the following critical stresses exist:

(a) the radial stresses at the inside diameter of the ring,
(b) the hoop stresses at the inside diameter of the ring,
(c) the axial hub stress at the surface of the junction with the ring.

In tapered-hub flanges sufficient reinforcement is added to the large side so that the critical axial stresses tend to move from the large to the small side of the hub or even into the shell. In loose flanges the maximum bending stresses occur always at the large end of the hub.

The stresses in a tapered-hub flange subjected to hydrostatic pressure only and to no bolt load were also analyzed. It was found that the maximum stresses in the hub, due to bending caused by the internal pressure, are as large as the direct tensile stress in the shell produced by the hydrostatic end force.

In the following, design formulas are given for the three critical stresses, which are applicable to the calculation of loose-ring flanges, flanges with hubs and tapered-hub flanges:

$$S_H = \frac{fM}{Lg_1^2} \tag{9}$$

$$S_R = \frac{\beta M}{Lt^2} \tag{10}$$

$$S_T = \frac{MY}{t^2} - ZS_R \tag{11}$$

where S_H = maximum axial stress at outer surface of hub or shell, lb. per sq. in.

S_R = radial stress at the inside diameter of the ring, in ring face next to hub, lb. per sq. in.

S_T = hoop stress coincident with S_R, lb. per sq. in.

$M = W \cdot a/B$.

W = total bolt load, lb.

a = lever arm, in.

A = outside diameter of flange, in.

B = inside diameter of ring, hub, and shell, in.

g_o = shell thickness, in.

g_1 = maximum hub thickness, in.

t = ring thickness, in.

f = ratio of S to axial stress at outer surface of hub where hub joins flange = stress-correction factor, Fig. 34.

$K = A/B$

F and V = factors for integral flanges, Fig. 35

F_L and V_L = factors for loose hubbed flanges, Fig. 35A

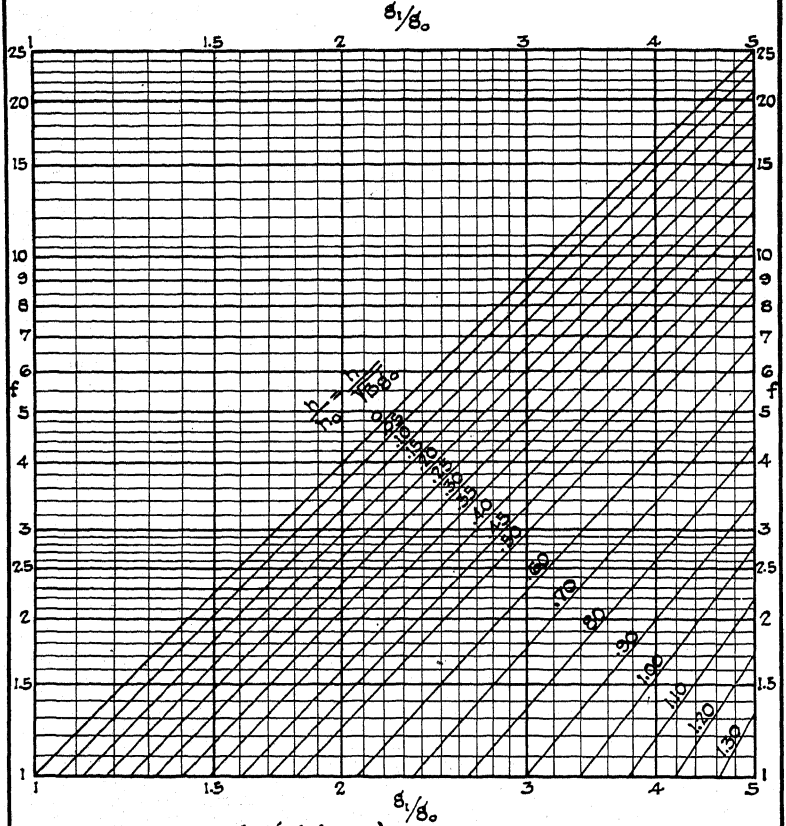
$$\beta = \frac{4}{3} te+1$$

$L = [(te+1)/T] - t^3/d$

$d = h_o g_o^2 \, U/V$ for integral flanges

$e = F/h_o$ for integral flanges

(substitute F_L and V_L for loose hubbed flanges)



f = 1(minimum)

f = 1 for hubs of uniform thickness $(\delta_1/\delta_0 = 1)$

f = 1 for loose hubbed flanges

Fig. 34. Values of "f" (Hub Stress Correction Factor) (Courtesy of Taylor Forge & Pipe Works).

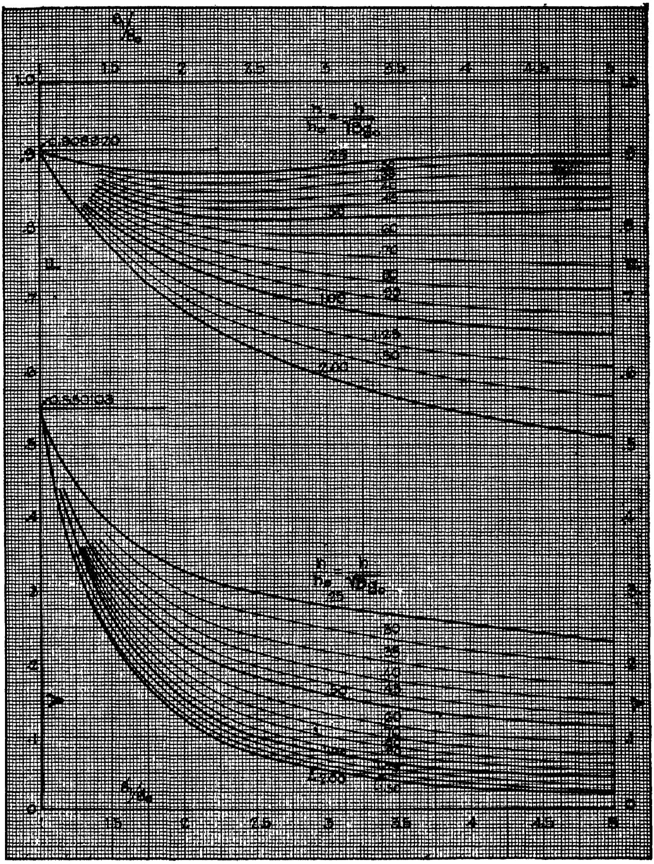


Fig. 35. Values of F and V (Integral Flange Factors) (Courtesy of Taylor Forge & Pipe Works).

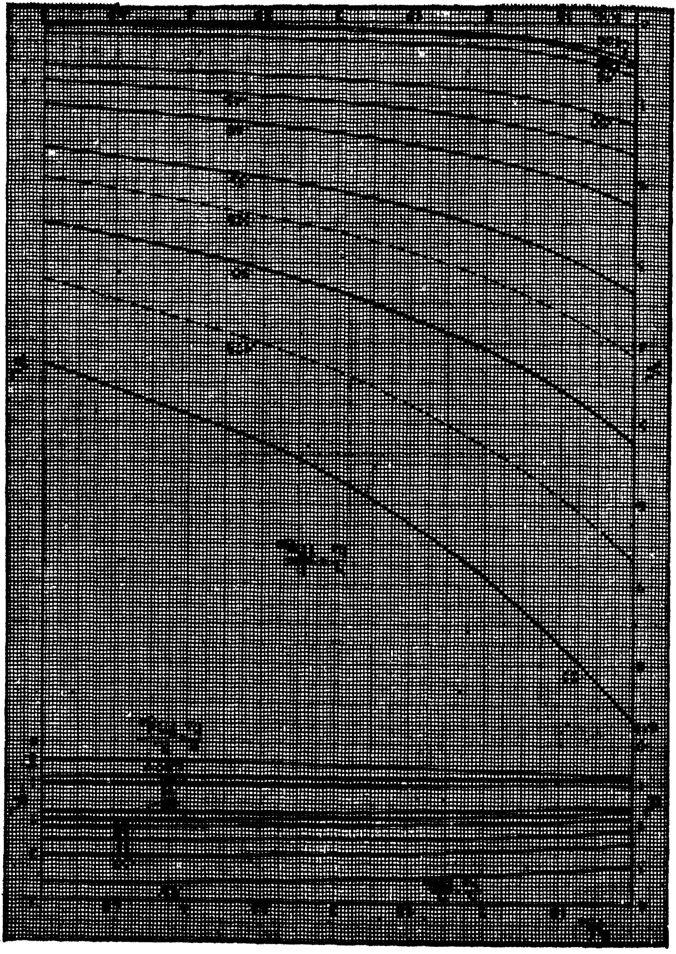


Fig. 35a. Values of F_L and V_L (Loose Hub Flange Factors) (Courtesy of Taylor Forge and Pipe Works).

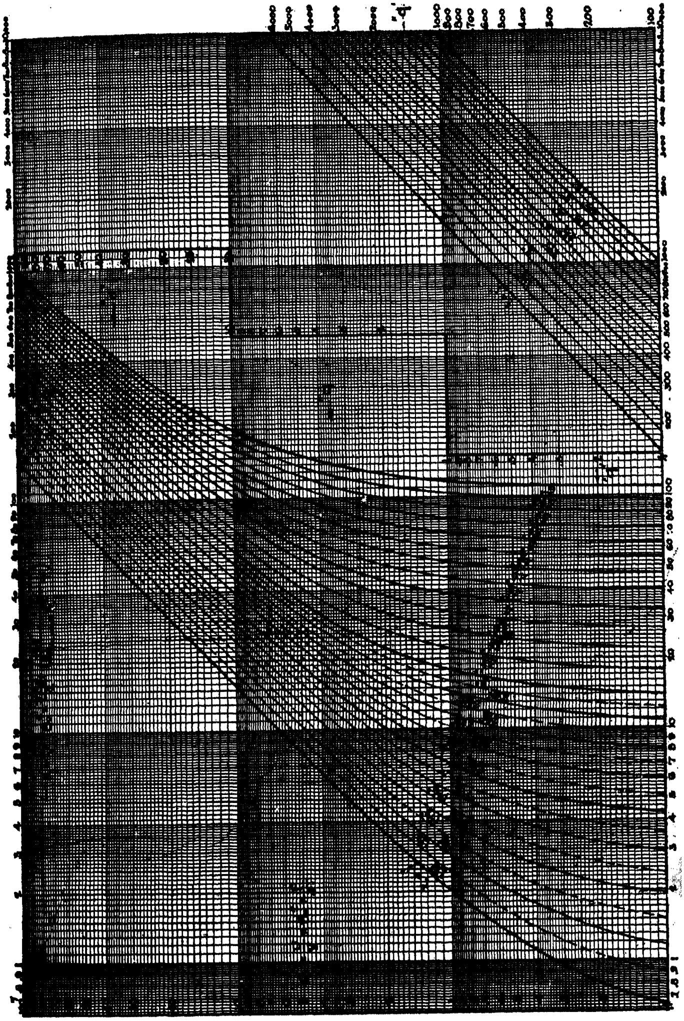


Fig. 35b. Values of t in terms of "a' and "g".
(Courtesy of Taylor Forge and Pipe Works).

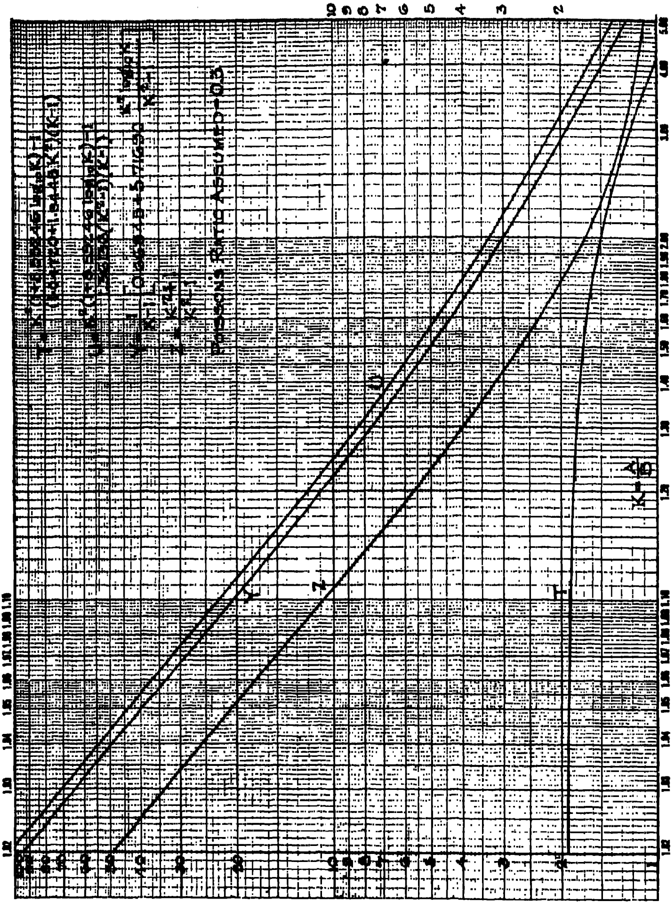


Fig. 36. Values of T, U, Y, Z. (Courtesy of Taylor Forge & Pipe Works).

$$T = \frac{3K^2\left(1 + 2\left(\frac{1 + \mu}{1 - \mu}\right) \cdot 2.3026 \log_{10}K\right) - 3}{\pi(K - 1)\left(1 + \frac{1 + \mu}{1 - \mu}K^2\right)}$$

$$U = \frac{3K^2\left(1 + 2\left(\frac{1 + \mu}{1 - \mu}\right) \cdot 2.3026 \log_{10}K\right) - 3}{\pi(1 + \mu)(K^2 - 1)(K - 1)}$$

$$Y = \frac{1}{K - 1}\left[\frac{3}{\pi}(1 - \mu) + \frac{6}{\pi}(1 + \mu)\frac{K^2}{K^2 - 1} \cdot 2.3026 \log_{10}K\right]$$

$$K^2 + 1$$

μ = Poisson's ratio (= 0.3 in design formulas).

Table 15. - Bolt Data for Flange Design
(All dimensions in inches.)

Bolt Size	Root Area		Over-Pulling Factor	Bolt Spacing		Min. Radial Distance	Edge Distance	Max. Fillet Rad.
	Std. Thr'd.	8 Thr'd.		Min.	Preferred			
$\frac{1}{2}$	.126	....	2.54	$1\frac{1}{4}$	3	$\frac{13}{16}$	$\frac{5}{8}$	$\frac{1}{4}$
$\frac{5}{8}$	.202	....	1.98	$1\frac{1}{2}$	3	$\frac{15}{16}$	$\frac{3}{4}$	$\frac{5}{16}$
$\frac{3}{4}$	.302	....	1.59	$1\frac{3}{4}$	3	$1\frac{1}{8}$	$\frac{13}{16}$	$\frac{3}{8}$
$\frac{7}{8}$	.419	....	1.33	$2\frac{1}{16}$	3	$1\frac{1}{4}$	$\frac{15}{16}$	$\frac{3}{8}$
1	.551	.551	1.16	$2\frac{1}{4}$	3	$1\frac{3}{8}$	$1\frac{1}{16}$	$\frac{7}{16}$
$1\frac{1}{8}$	.693	.728	....	$2\frac{1}{2}$	3	$1\frac{1}{2}$	$1\frac{1}{8}$	$\frac{7}{16}$
$1\frac{1}{4}$	.890	.929	....	$2\frac{13}{16}$	3	$1\frac{3}{4}$	$1\frac{1}{4}$	$\frac{9}{16}$
$1\frac{3}{8}$	1.054	1.155	....	$3\frac{1}{16}$	$3\frac{1}{16}$	$1\frac{7}{8}$	$1\frac{3}{8}$	$\frac{9}{16}$
$1\frac{1}{2}$	1.294	1.405	....	$3\frac{1}{4}$	$3\frac{1}{4}$	2	$1\frac{1}{2}$	$\frac{5}{8}$
$1\frac{5}{8}$	1.515	1.680	....	$3\frac{1}{2}$	$3\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{5}{8}$	$\frac{5}{8}$
$1\frac{3}{4}$	1.744	1.980	....	$3\frac{3}{4}$	$3\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{3}{4}$	$\frac{5}{8}$
$1\frac{7}{8}$	2.049	2.304	....	4	4	$2\frac{3}{8}$	$1\frac{7}{8}$	$\frac{5}{8}$
2	2.300	2.652	....	$4\frac{1}{4}$	$4\frac{1}{4}$	$2\frac{1}{2}$	2	$\frac{11}{16}$
$2\frac{1}{8}$	....	3.040	....	$4\frac{1}{2}$	$4\frac{1}{2}$	$2\frac{5}{8}$	$2\frac{1}{8}$	$\frac{11}{16}$
$2\frac{1}{4}$	3.020	3.423	....	$4\frac{3}{4}$	$4\frac{3}{4}$	$2\frac{3}{4}$	$2\frac{1}{4}$	$\frac{11}{16}$
$2\frac{1}{2}$	3.715	4.292	....	$5\frac{1}{4}$	$5\frac{1}{4}$	$3\frac{1}{16}$	$2\frac{3}{8}$	$\frac{13}{16}$
$2\frac{3}{4}$	4.618	5.259	....	$5\frac{3}{4}$	$5\frac{3}{4}$	$3\frac{3}{8}$	$2\frac{5}{8}$	$\frac{7}{8}$
3	5.428	6.324	....	$6\frac{1}{4}$	$6\frac{1}{4}$	$3\frac{5}{8}$	$2\frac{7}{8}$	$\frac{15}{16}$

Values of T. U. Y, and Z are obtained from Fig. 36, which has been reprinted from "Modern Flange Design," (ref. 9). Fig. 34 and 35 were reprinted from the same source. For loose ring flanges S_H and S_R equal zero, so that the hoop stress is $S_T = MY/t^2$, where $M = W\ a/B$.

Useful bolt data for different sizes of bolts are given in Table 15.

Revised A.S.M.E. Design of Flanges. - In a proposed revision of the A.S.M.E. Boiler Code (ref. 10), the Boiler code committee ruled, that the minimum required bolt W_m in pounds should be sufficient under operating conditions to resist the hydrostatic end force H in pounds exerted by the internal working pressure p upon the area bounded by the mean diameter of the gasket surface, and in addition maintain a predetermined compression load H_p on the gasket surface sufficient to assure a tight joint.

Note: Under the former rules, H was considered acting on the area bounded by the outside diameter of the gasket area.

In addition, the bolt load should be sufficient to initially yield the gasket without the presence of any internal pressure.

These conditions are expressed by the following conditions:

a) For operation:

$$W_m = H + H_p = 0.785\ G^2p + (2b \times 3.14\ Gmp) \qquad (12)$$

b) For gasket yielding:

$$W_m^1 = H_y = 3.14\ b\ G\ yr \qquad (13)$$

where G = mean diameter of gasket, inches

y = yield point of gasket, lbs. per sq. in. (table 15a)

b = effective gasket yielding width, inches (table 15b) (width to which unit load = y must be applied to set gasket)

2b = effective gasket pressure width, inches (width to which unit load = mp must be applied for a tight joint)

m = unit gasket compression factor (table 15a)

p = internal pressure, lbs. per sq. in.

$r = s_b/s_a$,

s_a = maximum allowable bolt stress at

atmospheric temperatures, lb. per sq. in
= 1.25 times the values given in Table A-2.

S_b = maximum allowable bolt stress at operating temperature, lb. per sq. in.,
= 1.25 times the values given in Table A-2.

H_p = $2\ b\ \pi\ G\ m\ p$ = pressure gasket load, lb.

H_y = $b\ \pi\ G\ y\ r$ = yield gasket load, lb.

In order to allow a direct comparison of the yielding and operating loading, the load required for yielding the gasket has been corrected by multiplication with the factor r.

Table 15a. - Gasket Factor m and Yield Point y.

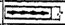
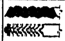
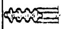
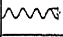
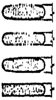
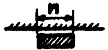
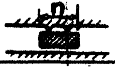
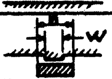
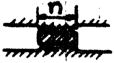
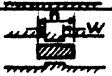
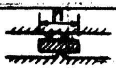
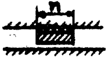
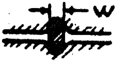
GASKET MATERIAL		GASKET FACTOR m*	YIELD POINT y	SKETCH AND NOTES
A. GUM RUBBER SHEET		.50	500	
B. CLOTH-INSERTED SOFT RUBBER, OR HARD RUBBER SHEET		.75	750	
C. CLOTH-INSERTED HARD RUBBER		1.00	1 000	
D. VEGETABLE FIBRE SHEET (Hemp or Jute)		1.50	2 000	
E. COMPRESSED ASBESTOS, OR ASBESTOS COMPOSITION		2.50	4 500	
F. WIRE MESH REINFORCED ASBESTOS		2.50	4 500	
G. CORRUGATED METAL, ASBESTOS INSERTED, OR SPIRAL-WOUND METAL, ASBESTOS FILLED		2.50	4 500	 Facing ①, Table B only.
H. CORRUGATED METAL JACKET, ASBESTOS FILLED		3.00	6 000	 Facing ①, Table B only.
J. CORRUGATED METAL	a) Copper	3.00	6 000	 Facing ①, Table B only.
	b) Monel c) Iron d) Soft Steel	3.25	7 000	
K. FLAT METAL JACKET, ASBESTOS FILLED	a) Aluminum b) Copper	3.25	7 000	
	c) Monel d) Iron e) Soft Steel	3.50	8 000	
	f) 4-6% Chrome g) 11-13% Chrome h) KA2S j) Type 316	3.75	9 000	
L. SOLID METAL	a) Soft aluminum	4.00	10 000	
	b) Soft Copper c) Admiralty	4.75	14 000	
	d) Iron e) Soft Steel f) Monel	5.50	18 000	
	g) 4-6% Chrome h) 11-13% Chrome	6.00	21 000	
	j) KA2S k) Type 316	6.50	24 500	

Table 15b. Effective Gasket Yielding Width b.

FACING	b	FACING	b
① 	$\frac{n}{2}$	⑤ 	$\frac{n}{3}$
② 	$\frac{n+w}{4}$	⑥ 	$\frac{n}{4}$
③ 	$\frac{n}{4}$	⑦ 	
④ 	$\frac{n}{3}$	⑧ 	$\frac{w}{4}$

The larger of the two values obtained by equations (12) and (13) represents the minimum bolt load W_m. The bolting is ordinarily selected to conform to the minimum requirements. But usually some excess results due to the custom of selecting the number of bolts in multiples of four and, particularly in low pressure designs, due to the necessity of maintaining a certain bolt spacing, in order to insure the tightness of the joint. The Actual Bolt Load W_a, pounds, then becomes

$$W_a = A_b \times S_b \qquad (14)$$

where A_b total cross-sectional area of bolts at root of thread or section of least diameter, sq. in.

The Design Bolt Load W, pounds, is then computed as the arithmetic mean of the minimum bolt load W_m and the actual bolt load W_a, or

$$W = (W_m + W_a)/2 \qquad (15)$$

Heretofore, the flange design had been based on the actual bolt load (see p. 105) which frequently produced excessive flange thicknesses. Previous design practice has been further modified by omitting the overpulling factor (Table 15), which used to be applied to the bolt sizes one inch and less. The design is thereby based on the desired working stress of the bolting material rather than on the stress produced by indiscriminate tightening. It is assumed that reasonable care will be taken to avoid overpulling of the bolts.

The Total Moment M_O is next computed as the sum of all the component moments acting upon the flange, or

$$Mo = M_D + M_T + M_G \tag{16}$$

The Unit Moment is then

$$M = M_O/B$$

The location of the forces acting upon the flanges and their respective lever arms are shown in Fig. 28. The moments are obtained in accordance with Table 15c. No allowance shall be made for any possible reduction of the lever arm due to cupping of the flanges.

The Flange Stresses are then computed by means of equations (9), (10) and (11). If S_{all} is the maximum allowable working stress of the flange material, the flange stresses shall remain in the following limits:

a) Integral flanges

$$S_H \leqq 1.5\ S_{all}$$

$$S_R \leqq S_{all}$$

$$S_T \leqq S_{all}$$

$$\frac{S_H + S_R}{2} \leqq S_{all}$$

$$\frac{S_H + S_T}{2} \leqq S_{all}$$

b) Ring flanges attached to nozzles and loose ring flanges

$$S_H,\ S_R,\ \text{and}\ S_T \leqq S_{all}$$

The flange stress equations presuppose that a certain flange thickness is known or has been assumed. If this is not the case, the flange thickness may be obtained directly from Fig. 35b. The coordinates of this figure are

$$A = \frac{1}{2}\, d \left(fM - \frac{1}{T} \right)$$

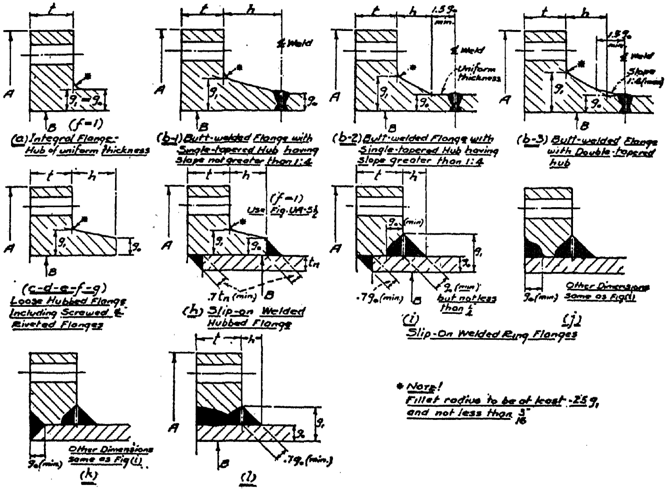
$$g = de/3T$$

Table 15c. - Formulas for Flange Moments

FLANGE LOADS	LEVER ARMS	FLANGE MOMENTS
Integral Flange (Fig. 28, type a,b,i,j,k, and l)		
$H_D = 0.785\ B^2 p$	$h_D = R + \frac{1}{2} g_1$	$M_D = H_D \times h_D$
$H_T = H - H_D$	$h_T = \frac{1}{2}(R+g_1+h_G)$	$M_T = H_T \times h_T$
$H_G = W - H$	$h_G = \frac{1}{2}(C-G)$	$M_G = H_G \times h_G$
Slip-on and Screwed Flange (Type c,e,g,h)		
$H_D = 0.785\ B^2 p$	$h_D = \frac{1}{2}(C-B)$	$M_D = H_D \times h_D$
$H_T = H - H_D$	$h_T = \frac{1}{2}(h_D+h_G)$	$M_T = H_T \times h_T$
$H_G = W - H$	$h_G = \frac{1}{2}(C-G)$	$M_G = H_G \times h_G$
Loose Flange (Type d and e)		
$H_D = 0.785\ B^2 p$	$h_D = \frac{1}{2}(C-B)$	$M_D = H_D \times h_D$
$H_T = H - H_D$	$h_T = \frac{1}{2}(C-G)$	$M_T = H_T \times h_T$
$H_G = W - H$	$h_G = \frac{1}{2}(C-G)$	$M_G = H_G \times h_T$
Loose Flange (Type f)		
W	h_W	$M_O = W \times h_W$

If the hub stress governs, the value obtained from Fig. 35b will be correct. In cases where the flange stress is the ruling factor, it affords a first step, and another value of t has to be assumed, which gives stresses within the limits of the allowable stresses.

Recommended proportions of flanges and requirements for taper of hubs and location of welds are shown in **Fig. 36a**.



(a) Integral Flange: Hub of uniform thickness

(b-1) Butt-welded Flange with Single-tapered Hub having Slope not greater than 1:4

(b-2) Butt-welded Flange with Single-tapered Hub having Slope greater than 1:4

(b-3) Butt-welded Flange with Double-tapered hub

(c-d-e-f-g) Loose Hubbed Flange Including Screwed & Riveted Flanges

(h) Slip-on Welded Hubbed Flange

(i) (j) Slip-On Welded Ring Flanges

(k)

(l)

* Note! Fillet radius to be at least .25 g_1 and not less than 3/16"

Fig. 36a. Dimensions and Proportions of Flanges

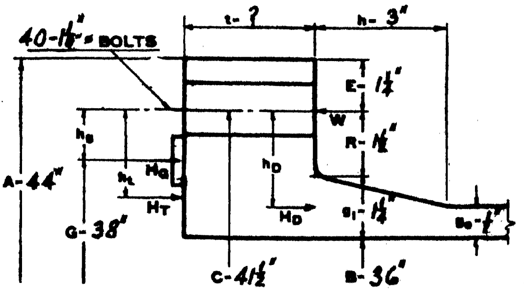


Fig. 36b. Design of 36 "Dia. Flange.

Example. Given: 36" I.D. welding neck flange as per Fig. 36b, operating pressure 300 lb. per sq. in., operating temperature 500°F, 37" I.D. x 39" O.D. x $\frac{1}{4}$" thick asbestos composition gasket, flange material: A.S.T.M. specification A-105, Grade 1, allowable flange stress 15000 lb. per sq. in., allowable hub stress S_H = 22500 lb. per sq. in., bolting material: A.S.T.M. specification A-96, Grade A, allowable bolt stress 16250 lb. per sq. in. (allowing a factor of safety = 4).

Required: Thickness of the flange.

Solution: Assume 40 - 1 1/8" dia. bolts, $h = 3$, $g_1 = 1\frac{1}{4}$, minimum $R = 1\frac{1}{2}$ (from Table 15) gives $C = 41\frac{1}{2}$, $E = 1\frac{1}{4}$, $A = 44$, $b = 0.5$ (from Table 15a), $y = 4500$ and $m = 2.50$ (from Table 15b).

Calculate $H = 340000$, $H_p = 90000$, $H_y = 270000$, $W_m = 340000 + 90000 = 430000$, $W_m' = 270000$, $W_a = 450000a$, $W = \frac{1}{2}$ $(430000 + 450000) = 440000$.

Flange moments (see Table 15c): $M_D = 306000 \text{ x } 2.125 = 651000$, $M_T = 34000 \text{ x } 2.25 = 76500$, $M_G = 100000 \text{ x } 1.75 = 175000$, $M_0 = 902500$, $M = 25200$.

Shape constants: $K = 44/36 = 1.22$, $g_1/g_0 = 2.5$, $h/h_0 = 0.71$, from Fig. 34: $f = 1.2$, from Fig. 35: $F = 0.77$, and $V = 0.142$, from Fig. 36: $T = 1.82$, $Z = 5.2$, $Y = 10.0$, and $U = 11.0$.

Calculate $d = 82.1$, $e = 0.1815$, $a = 12.77$ (for $S_H = 22500$), $g = 2.73$, enter Fig. 35b with a and g, and obtain $t = 2.125$. Check flange stresses: $S_R = S_H\beta\, g_1{}^2/Lt^2 = 9800$ (substituting L from eq. 9 and in eq. 10), $S_T = 4900$ (from eq. 11), $\frac{1}{2}(S_H + S_R) = 16150$. The last stress is excessive. Another value of $t = 2.375$ is, therefore, assumed and the stresses are checked again: $L = 0.951$, $S_H = 20400$, $S_R = 7280$ (from eq. 10), $S_T = 7100$, $\frac{1}{2}(S_H + S_R) = 13840$, and $\frac{1}{2}(S_H + S_T) = 13750$.

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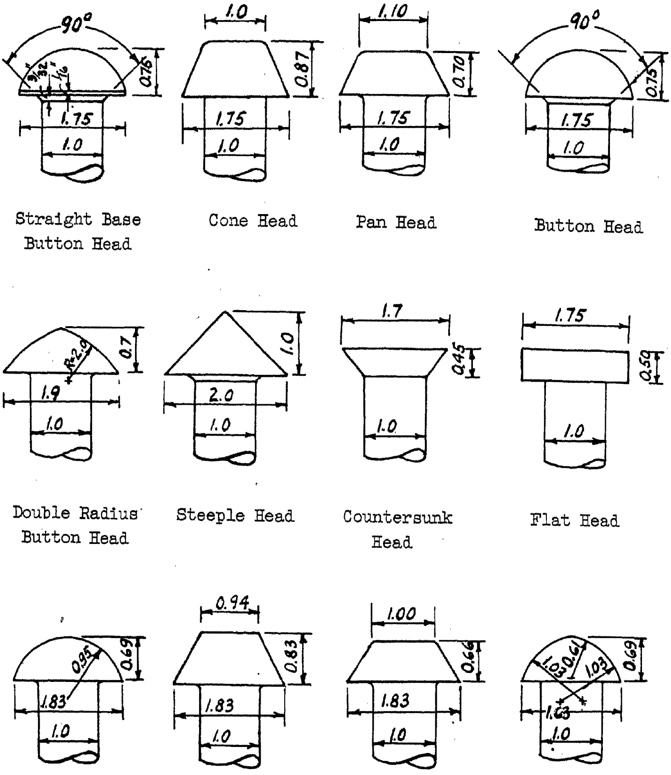
Chapter XIV

RIVETED JOINTS

Methods of Riveting. - Riveting is done cold or hot, and by hand or by hydraulic or pneumatic machines. In tank work rivets up to 3/8" diameter are driven cold. In joints which are subjected to shock or oscillating forces, cold riveting is sometimes applied for larger rivet diameters. Rivets in boilers are always driven hot. A rivet driven hot shrinks in cooling and pinches the riveted parts, thereby causing a frictional resistance between them. In some countries (Germany and France) the design is based upon the frictional resistance. In American and English practice, however, the tensile and shearing strength of the plates and rivets is used as basis for the design. Hand riveting can be used up to approximately one inch rivet diameter. But machine riveting is generally preferred, as the pressure of the machine forces the rivet shank to fill the rivet hole completely.

Rivet Holes. - The rivet holes in plates may be drilled or punched or prepunched and reamed. When drilled, which is preferable, the holes are drilled 1/16 in. larger than the rivet diameter so that the hot rivet can easily be entered. If the hole is to be reamed, it is usually punched to a diameter 1/8 or 3/16 in. less than the finished diameter and reamed to the proper size after the plates are assembled.

Rivet Heads and Lengths. - The form of the head varies, but is generally spherical or conical. The apex of the cone is usually cut off so that the head is shaped more like a frustrum. A number of different types of rivet heads and customary proportions are shown in Fig. 37 which has been redrawn from Fig. U-1 of the A.S.M.E. Code. Since countersunk rivets weaken the



Straight Base Button Head

Cone Head

Pan Head

Button Head

Double Radius Button Head

Steeple Head

Countersunk Head

Flat Head

Button Head

Cone Head

Pan Head

High Button Head

Fig. 37. Standard Proportions for Rivet Heads (A.S.A. Standard B18.4 - 1937)

plate they should be used only where projecting heads are objectionable. In order to form the head and fill the clearance between the rivet and the rivet hole, the rivet should have a length in excess of the thickness of the plate equal to about three-fourths the diameter for the countersunk head, and from 1.3 to 1.7 times the diameter for ordinary riveting. (See table 16.)

Caulking. - Plates less than 3/16" thick cannot be caulked. The joints are sealed in this case by inserting cord, red lead, etc. The seams of boilers have to be caulked. The caulking edges of plates should be beveled to an angle not sharper than 70 deg. to the plane of the plate.

Plate Thickness. - The plate thickness is calculated by means of the formulas given for pressure vessels or open tanks, using the proper efficiency value corresponding to the type of joint selected.

Types of Joint. - Plates may be joined by means of a lap-joint, a butt-joint, or a combination lap-joint. The latter consists of a cover plate or butt strap inside or outside the lap, and three rows of rivets, the middle one of which passes through the two plates and the cover (Fig. 38).

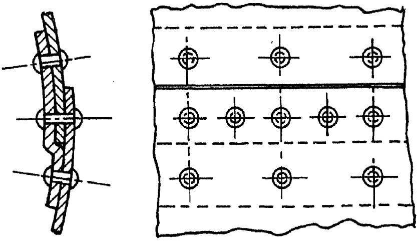
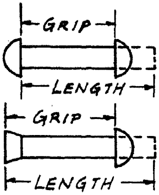


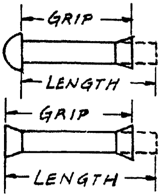
Fig. 38

Lap joints should not be used when the required thickness of the plate exceeds one-half inch on account of the bending stresses produced in this type of joint.

According to the number of rows single-riveted, double-riveted, etc., joints are distinguished. Single-riveting means one row of rivets in a lap joint, or one row on each side of a butt joint; double-riveting means two rows of rivets in a lap-joint, or two rows on each

Table 16. - Length of Rivets Required for Various Grips Including Amount Necessary to Form One Head





Grip in Inches	Diameter of Rivet							Diameter of Rivet						
	1/4"	3/8"	1/2"	5/8"	3/4"	7/8"	1"	1/4"	3/8"	1/2"	5/8"	3/4"	7/8"	1"
	Length of Rivet							Length of Rivet						
1/2	1	1 1/4	1 5/8	1 3/4	1 7/8	2	2 1/8	7/8	1	1	1	1 1/8	1 1/4	1 1/4
5/8	1 1/8	1 3/8	1 3/4	1 7/8	2	2 1/8	2 1/4	1	1 1/8	1 1/8	1 1/4	1 1/4	1 3/8	1 3/8
3/4	1 1/4	1 1/2	1 7/8	2	2 1/8	2 1/4	2 3/8	1 1/8	1 1/4	1 3/8	1 3/8	1 3/8	1 1/2	1 1/2
7/8	1 3/8	1 5/8	2	2 1/8	2 1/4	2 3/8	2 1/2	1 1/4	1 3/8	1 1/2	1 1/2	1 1/2	1 5/8	1 5/8
1	1 1/2	1 3/4	2 1/4	2 3/8	2 3/8	2 1/2	2 5/8	1 3/8	1 1/2	1 5/8	1 5/8	1 5/8	1 3/4	1 3/4
1 1/8	1 5/8	1 7/8	2 3/8	2 1/2	2 1/2	2 5/8	2 3/4	1 1/2	1 5/8	1 3/4	1 3/4	1 7/8	1 7/8	1 7/8
1 1/4	1 3/4	2	2 1/2	2 5/8	2 5/8	2 3/4	2 7/8	1 5/8	1 3/4	2	2	2	2	2
1 3/8	1 7/8	2 1/8	2 5/8	2 3/4	2 3/4	2 7/8	3	1 3/4	1 7/8	2 1/8	2 1/8	2 1/8	2 1/4	2 1/4
1 1/2	2	2 1/4	2 7/8	2 7/8	3	3 1/8	3 1/4	1 7/8	2	2 1/4	2 1/4	2 1/4	2 3/8	2 3/8
1 5/8	2 1/8	2 3/8	3	3	3 1/8	3 1/4	3 3/8	2	2 1/8	2 3/8	2 3/8	2 3/8	2 1/2	2 1/2
1 3/4	2 1/4	2 1/2	3 1/8	3 1/4	3 1/4	3 1/2	3 1/2	2 1/8	2 1/4	2 5/8	2 5/8	2 5/8	2 5/8	2 5/8
1 7/8	2 3/8	2 5/8	3 1/4	3 3/8	3 3/8	3 5/8	3 5/8	2 1/4	2 3/8	2 3/4	2 3/4	2 3/4	2 3/4	2 3/4
2	2 1/2	2 3/4	3 1/2	3 1/2	3 5/8	3 3/4	3 3/4	2 3/8	2 1/2	2 7/8	2 7/8	2 7/8	2 7/8	2 7/8
2 1/8	2 5/8	2 7/8	3 5/8	3 5/8	3 3/4	3 7/8	3 7/8	2 1/2	2 5/8	3 1/8	3	3	3	3
2 1/4	2 3/4	3	3 3/4	3 7/8	3 7/8	4	4 1/8	2 5/8	2 3/4	3 1/4	3 1/8	3 1/8	3 1/8	3 1/4
2 3/8	2 7/8	3 1/8	4	4	4	4 1/8	4 1/4	2 3/4	2 7/8	3 3/8	3 3/8	3 3/8	3 3/8	3 3/8
2 1/2	3	3 1/4	4 1/8	4 1/8	4 1/8	4 1/4	4 3/8	2 7/8	3	3 1/2	3 1/2	3 1/2	3 1/2	3 1/2
2 5/8	3 1/8	3 1/2	4 1/4	4 1/4	4 1/4	4 3/8	4 1/2	3	3 1/4	3 3/4	3 5/8	3 5/8	3 5/8	3 5/8
2 3/4	3 1/4	3 5/8	4 3/8	4 3/8	4 3/8	4 1/2	4 5/8	3 1/8	3 3/8	3 7/8	3 3/4	3 3/4	3 3/4	3 3/4
2 7/8	3 3/8	3 3/4	4 5/8	4 5/8	4 5/8	4 5/8	4 3/4	3 1/4	3 1/2	4	3 7/8	3 7/8	3 7/8	3 7/8
3	3 1/2	3 7/8		4 3/4	4 3/4	4 7/8	4 7/8	3 3/8	3 5/8		4 1/8	4 1/8	4 1/8	4 1/8
3 1/8	3 5/8	4		4 7/8	4 7/8	5	5 1/8	3 1/2	3 3/4		4 1/4	4 1/4	4 1/4	4 1/4
3 1/4	3 3/4	4 1/8		5	5	5 1/8	5 1/4	3 5/8	3 7/8		4 3/8	4 3/8	4 3/8	4 3/8
3 3/8	3 7/8	4 1/4		5 1/8	5 1/8	5 1/4	5 3/8	3 3/4	4		4 1/2	4 1/2	4 1/2	4 1/2
3 1/2	4	4 3/8		5 3/8	5 3/8	5 3/8	5 1/2	3 7/8	4 1/8		4 5/8	4 5/8	4 5/8	4 5/8
3 5/8	4 1/8	4 1/2		5 1/2	5 1/2	5 1/2	5 5/8	4	4 1/4		4 3/4	4 3/4	4 3/4	4 3/4
3 3/4	4 1/4	4 5/8		5 5/8	5 5/8	5 5/8	5 3/4	4 1/8	4 3/8		5	5	5	5
3 7/8	4 3/8	4 3/4		5 3/4	5 3/4	5 3/4	5 7/8	4 1/4	4 1/2		5 1/8	5 1/8	5 1/8	5 1/8
4	4 1/2	4 7/8			5 7/8	6	6	4 3/8	4 5/8			5 1/4	5 1/4	5 1/4
4 1/8	4 5/8	5			6	6 1/8	6 1/4	4 1/2	4 3/4			5 3/8	5 3/8	5 3/8
4 1/4	4 3/4	5 1/8			6 1/8	6 1/4	6 3/8	4 5/8	4 7/8			5 1/2	5 1/2	5 1/2
4 3/8	4 7/8	5 1/4			6 3/8	6 1/2	6 1/2	4 3/4	5			5 5/8	5 5/8	5 5/8
4 1/2	5	5 3/8			6 1/2	6 5/8	6 5/8	4 7/8	5 1/8			5 3/4	5 3/4	5 3/4
4 5/8	5 1/8	5 1/2			6 5/8	6 3/4	6 3/4	5	5 1/4			6	6	6
4 3/4	5 1/4	5 5/8			6 3/4	6 7/8	6 7/8	5 1/8	5 3/8			6 1/8	6 1/8	6 1/8
4 7/8					6 7/8	7	7					6 1/4	6 1/4	6 1/4
5						7 1/8	7 1/8						6 3/8	6 3/8
5 1/8						7 1/4	7 1/4						6 1/2	6 1/2
5 1/4						7 3/8	7 3/8						6 5/8	6 5/8
5 3/8						7 5/8	7 5/8						6 3/4	6 3/4
5 1/2						7 3/4	7 3/4						6 7/8	6 7/8
5 5/8						7 7/8	7 7/8						7	7
5 3/4						8	8						7 1/4	7 1/4
5 7/8						8 1/8	8 1/8						7 3/8	7 3/8

side of the joint in a butt-joint.

Butt joints are in general more desirable and in most cases the higher labor cost will be offset by the reduction in plate thickness obtained through the greater joint efficiency.

The stresses in circumferential joints of pressure vessels are only half as large as in longitudinal joints and, therefore, single-riveted joints are used for them when double-riveted joints are used for the longitudinal joints.

Rivet Diameter. - Approximate values of rivet diameter d in proportion to plate thickness t are

Single-riveted joints: $d = 1.4\sqrt{t}$
Multiple riveted joints: $d = 1.2\sqrt{t}$

Rivet Spacing. - The center-to-center distance or pitch of rivets has to be large enough to provide access for the setting tool, but should not be too large so that it would impair the tightness of the joint. It is customary practice to limit the pitch to the following proportions:

Minimum distance:	Pitch $P = 3d$
Maximum distance:	8t, for one row
	9t, for two or more rows
	10t, for water tanks

where d = diameter of rivet, in.
t = thickness of plate, in.

The distance from the center line of rivets to the edge of the plate shall not be less than 1-1/2 and not more than 1-3/4 times the diameter of the rivet holes. The distance between center lines of any two adjacent rows of rivets or back pitch "A" (see Fig. 39) vertical to the joint shall have the following minimum values:

$$A = 1\text{-}3/4d + 0.1(P - 4d); \text{ for } \frac{P}{d} \text{ larger than } 4$$

where d = diameter of rivet holes, in.

pitch of rivets in outer row where a rivet in the inner row comes midway between two rivets in the outer row, in.

pitch of rivets in the outer row less pitch of rivets in the inner row where two rivets in the inner row come between two rivets in the outer row, in.

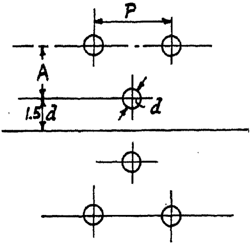


Fig. 39

Strength and Efficiency of Riveted Joints. - The strength of a riveted joint is usually expressed by its efficiency, i.e., the ratio which the strength of the joint bears to the strength of the solid plate. A failure of the joint may be caused by shearing the rivets, tearing the plate between the rivets, crushing the rivets or plate, or by a combination of these causes. In order to determine the strength of a joint, the breaking strength of a unit section is calculated for all possible modes of failure. The lowest value represents the actual strength of the section. The efficiency of the joint is then obtained by dividing the lowest value by the breaking strength of the solid plate for a length equal to that of the section considered.

In the following the stress equations are given for several types of joints as presented in the A.S.M.E. Boiler Code:

Let, TS = tensile strength stamped on plate, lb. per sq. in.
t = thickness of plate, in.
b = thickness of butt strap, in.
P = pitch of rivets, in., on row having greatest pitch
d = diameter of rivet after driving, in. = diameter of rivet hole
a = cross-sectional area of rivet after driving, sq. in.
s = shearing strength of rivet in single shear, lb. per sq. in.

S = shearing strength of rivet in double shear, lb. per sq. in.
c = crushing strength of mild steel, lb. per sq. in.
n = number of rivets in single shear in a unit length of joint
N = number of rivets in double shear in a unit length of joint.

A. Lap-joint, single-riveted, Fig. 40.
(1) Strength of solid plate = P x t x TS
(2) Strength of plate between rivet holes = (P - d)t x TS
(3) Shearing strength of one rivet in single shear $= n \times s \times a = \frac{\pi d^2}{4} s$
(4) Crushing strength of plate in front of one rivet = d x t x c.

B. Lap-joint, double-riveted, Fig. 41.
(1) Strength of solid plate = P x t x TS
(2) Strength of plate between rivet holes = (P - d)t x TS
(3) Shearing strength of two rivets in single shear = n x s x a
(4) Crushing strength of plate in front of two rivets = n x d x t x c

C. Butt joint with double strap, double-riveted, Fig. 42.
(1) Strength of solid plate = P x t x TS
(2) Strength of plate between rivet holes in the outer row = (P - d)t x TS
(3) Shearing strength of two rivets in double shear, plus the shearing strength of one rivet in single shear = N x S x a + n x s x a
(4) Strength of plate between rivet holes in the second row, plus the shearing strength of one rivet in single shear in the outer row = (P - 2d)t x TS + n x s x a
(5) Strength of plate between rivet holes in the second row, plus the crushing strength of butt strap in front of one rivet in the outer row = (P - 2d)t x TS + d x b x c
(6) Crushing strength of plate in front of two rivets, plus the crushing strength of butt strap in front of one rivet = N x d x t x c + n x d x b x c

(7) Crushing strength of plate in front of two rivets, plus the shearing strength of one rivet in single shear = N x d x t x c + n x s x a

D. Butt-joint with double strap, triple-riveted, Fig. 43.

(1) Strength of solid plate = P x t x TS

(2) Strength of plate between rivet holes in the outer row = (P - d)t x TS

(3) Shearing strength of four rivets in double shear, plus the shearing strength of one rivet in single shear = (P - 2d)t x TS + n x s x a

(4) Strength of plate between rivet holes in the second row, plus the shearing strength of one rivet in single shear in the outer row
= (P - 2d)t x TS + n x s x a

(5) Strength of plate between rivet holes in the second row, plus the crushing strength of butt strap in front of one rivet in the outer row
= (P - 2d)t x TS + d x b x c

(6) Crushing strength of plate in front of four rivets, plus the crushing strength of butt strap in front of one rivet = N x d x t x c + n x d x b x c

(7) Crushing strength of plate in front of four rivets, plus the shearing strength of one rivet in single shear = N x d x t x c + n x s x a.

E. Butt joint with double strap, quadruple-riveted, Fig. 44.

(1) Strength of solid plate = P x t x TS

(2) Strength of plate between rivet holes in the outer row = (P - d)t x TS

(3) Shearing strength of eight rivets in double shear, plus the shearing strength of three rivets in single shear = N x S x a + n x s x a

(4) Strength of plate between rivet holes in the second row, plus the shearing strength of one rivet in single shear in the outer row
= (P - 2d)t x TS + 1 x s x a

(5) Strength of plate between rivet holes in the third row, plus shearing strength of two rivets in the second row in single shear and one rivet in single shear in the outer row
= (P - 4d)t x TS + n x s x a

(6) Strength of plate between rivet holes in the second row, plus the crushing strength of butt strap in front of one rivet in the outer row
$= (P - 2d)t \times TS + d \times b \times c$

(7) Strength of plate between rivet holes in the third row, plus the crushing strength of butt strap in front of two rivets in the second row and one rivet in the outer row
$= (P - 4d)t \times TS + n \times d \times b \times c$

(8) Crushing strength of plate in front of eight rivets, plus the crushing strength of butt strap in front of three rivets
$= N \times d \times t \times c + n \times d \times b \times c$

(9) Crushing strength of plate in front of eight rivets, plus the shearing strength of two rivets in the second row and one rivet in the outer row, in single shear $= N \times d \times t \times c + n \times s \times a$

Working Stresses.

(a) Plate:
Tensile strength 55,000 lbs. per sq. in.
Crushing strength 95,000 lbs. per sq. in.

(b) Rivets:
Ultimate strength of rivets in shear in lb. per sq. in.

Iron rivets in single shear	38,000
Iron rivets in double shear	76,000
Steel rivets in single shear	44,000
Steel rivets in double shear	88,000
Alloy-steel rivets, A.S.M.E. Spec. S-28, Grade A, in single shear	60,000
Alloy-steel rivets, A.S.M.E. Spec. S-28, in double shear	120,000

In figuring the stresses the nominal diameter of the finished holes is taken as the shank diameter of the rivets.

Joint efficiencies and dimensions for several types of joints, as recommended by the Hartford Steam Boiler Inspection and Insurance Company, are presented in the following tables:

Table 18, Lap joint, single riveted
Table 19, Lap joint, double riveted

Table 20, Butt and double strap joint, double riveted

Table 21, Butt and double strap joint, triple riveted

Table 22, Butt and double strap joint, quadruple riveted

Table 23, Lap-riveted joints for girth seams.

REFERENCES

1. A.S.M.E. Code for Power Boilers, Edition 1937.
2. The Hartford Steam Boiler Inspection and Insurance Company, The Boiler Book, Hartford, Conn., 1928.

Table 18. - Lap Joint--Single Riveted (Fig. 40)
(Recommended by Hartford S.B.I. & I. Co.)

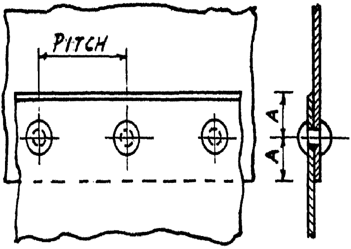


Fig. 40

Thickness of Plate, Inches	Diameter of Cold Rivet, Inches	Diameter of Rivet Holes, Inches	Pitch Inches	A, Inches	Efficiency Per Cent
$\frac{1}{4}$	$\frac{5}{8}$	$\frac{11}{16}$	$1\frac{3}{4}$	$1\frac{1}{16}$	60.7
$\frac{9}{32}$	$\frac{5}{8}$	$\frac{11}{16}$	$1\frac{3}{4}$	$1\frac{1}{16}$	60.3
$\frac{5}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	2	$1\frac{1}{4}$	59.3
$\frac{11}{32}$	$\frac{3}{4}$	$\frac{13}{16}$	2	$1\frac{1}{4}$	59.4
$\frac{3}{8}$	$\frac{7}{8}$	$\frac{15}{16}$	$2\frac{1}{4}$	$1\frac{7}{16}$	58.3
$\frac{13}{32}$	$\frac{7}{8}$	$\frac{15}{16}$	$2\frac{1}{4}$	$1\frac{7}{16}$	58.3
$\frac{7}{16}$	1	$1\frac{1}{16}$	$2\frac{1}{2}$	$1\frac{5}{8}$	57.5
$\frac{15}{32}$	1	$1\frac{1}{16}$	$2\frac{1}{2}$	$1\frac{5}{8}$	57.5
$\frac{1}{2}$	1	$1\frac{1}{16}$	$2\frac{1}{2}$	$1\frac{5}{8}$	56.7

Table 19. - Lap Joint--Double Riveted (Fig. 41)
(Recommended by Hartford S.B.I. & I. Co.)

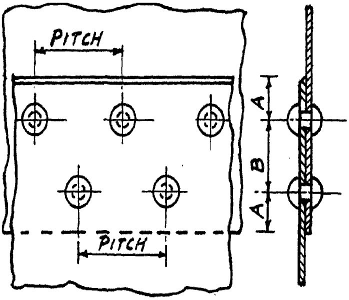


Fig. 41

Thick-ness of Plate, Inches	Diameter of Cold Rivet, Inches	Diameter of Rivet Holes, Inches	Pitch, Inches	A, Inches	B, Inches	Efficiency, Per Cent
$\frac{1}{4}$	$\frac{5}{8}$	$\frac{11}{16}$	$2\frac{1}{4}$	$1\frac{1}{16}$	$1\frac{3}{4}$	69.5
$\frac{9}{32}$	$\frac{5}{8}$	$\frac{11}{16}$	$2\frac{1}{4}$	$1\frac{1}{16}$	$1\frac{3}{4}$	69.5
$\frac{5}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$2\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{7}{8}$	69.1
$\frac{11}{32}$	$\frac{3}{4}$	$\frac{13}{16}$	$2\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{7}{8}$	69.1
$\frac{3}{8}$	$\frac{7}{8}$	$\frac{15}{16}$	3	$1\frac{7}{16}$	2	68.9
$\frac{13}{32}$	$\frac{7}{8}$	$\frac{15}{16}$	3	$1\frac{7}{16}$	2	68.9
$\frac{7}{16}$	1	$1\frac{1}{16}$	$3\frac{3}{8}$	$1\frac{5}{8}$	$2\frac{1}{8}$	68.5
$\frac{15}{32}$	1	$1\frac{1}{16}$	$3\frac{3}{8}$	$1\frac{5}{8}$	$2\frac{1}{8}$	68.5
$\frac{1}{2}$	1	$1\frac{1}{16}$	$3\frac{3}{8}$	$1\frac{5}{8}$	$2\frac{1}{8}$	68.5

Table 20. - Butt Joints--Double Riveted (Fig. 42)
(Recommended by Hartford S.B.I. & I. Co.)

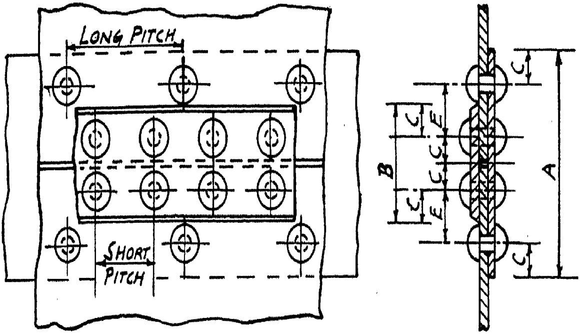
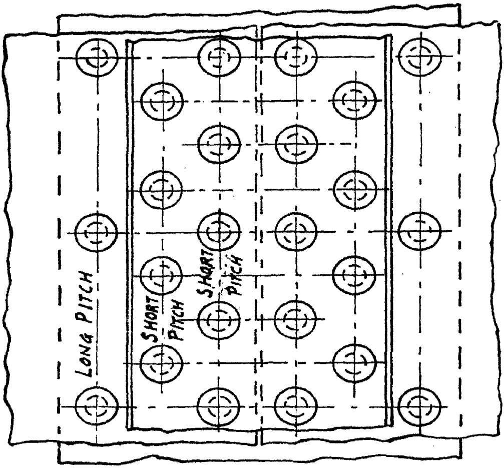


Fig. 42

Thickness of Plate, Inches	Thickness of Straps, Inches	Diameter of Cold Rivet, Inches	Diameter of Rivet Hole, Inches	Long Pitch, Inches	Short Pitch, Inches	A Inches	B Inches	C Inches	E Inches	Efficiency Per Cent
1/4	1/4	5/8	11/16	4	2	8 1/2	4 1/4	1 1/16	2 1/8	82.8
9/32	1/4	5/8	11/16	4	2	8 1/2	4 1/4	1 1/16	2 1/8	82.8
5/16	9/32	3/4	13/16	4 1/2	2 1/4	9 7/8	5	1 1/4	2 7/16	81.9
11/32	9/32	3/4	13/16	4 1/2	2 1/4	9 7/8	5	1 1/4	2 7/16	81.9
3/8	5/16	3/4	13/16	4 1/2	2 1/4	9 7/8	5	1 1/4	2 7/16	81.9
13/32	5/16	3/4	13/16	4 1/2	2 1/4	9 7/8	5	1 1/4	2 7/16	81.9
7/16	3/8	7/8	15/16	5	2 1/2	11 1/4	5 3/4	1 7/16	2 3/4	81.3
15/32	3/8	7/8	15/16	5	2 1/2	11 1/4	5 3/4	1 7/16	2 3/4	81.3
1/2	7/16	7/8	15/16	5	2 1/2	11 1/4	5 3/4	1 7/16	2 3/4	81.3



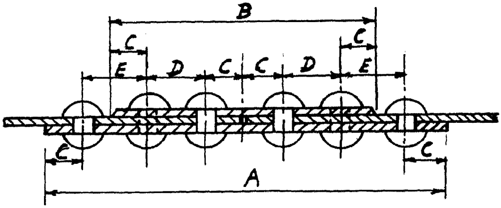


Fig. 43. Butt Joint--Triple Riveted
(Recommended by Hartford S.B.I. I. Co.)

Table 21. - Butt Joints--Triple Riveted (Fig. 43)
(Recommended by Hartford S.B.I. & I. Co.)

Thickness of Plate, Inches	Thickness of Straps, Inches	Diameter of Cold Rivet, Inches	Diameter of Rivet Hole, Inches	Long Pitch, Inches	Short Pitch, Inches	A Inches	B Inches	C Inches	D Inches	E Inches	Efficiency Per Cent
$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{8}$	$\frac{11}{16}$	$5\frac{1}{2}$	$2\frac{3}{4}$	12	$7\frac{3}{4}$	$1\frac{1}{16}$	$1\frac{3}{4}$	$2\frac{1}{8}$	87.5
$\frac{9}{32}$	$\frac{1}{4}$	$\frac{5}{8}$	$\frac{11}{16}$	$5\frac{1}{2}$	$2\frac{3}{4}$	12	$7\frac{3}{4}$	$1\frac{1}{16}$	$1\frac{3}{4}$	$2\frac{1}{8}$	87.5
$\frac{5}{16}$	$\frac{9}{32}$	$\frac{3}{4}$	$\frac{13}{16}$	$6\frac{1}{2}$	$3\frac{1}{4}$	$13\frac{5}{8}$	$8\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{7}{8}$	$2\frac{7}{16}$	87.5
$\frac{11}{32}$	$\frac{9}{32}$	$\frac{3}{4}$	$\frac{13}{16}$	$6\frac{1}{2}$	$3\frac{1}{4}$	$13\frac{5}{8}$	$8\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{7}{8}$	$2\frac{7}{16}$	87.5
$\frac{3}{8}$	$\frac{5}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	7	$3\frac{1}{2}$	$13\frac{5}{8}$	$8\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{7}{8}$	$2\frac{7}{16}$	88.4
$\frac{13}{32}$	$\frac{5}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	7	$3\frac{1}{2}$	$13\frac{5}{8}$	$8\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{7}{8}$	$2\frac{7}{16}$	88.4
$\frac{7}{16}$	$\frac{3}{8}$	$\frac{7}{8}$	$\frac{15}{16}$	$7\frac{3}{4}$	$3\frac{7}{8}$	$15\frac{1}{4}$	$9\frac{3}{4}$	$1\frac{7}{16}$	2	$2\frac{3}{4}$	87.9
$\frac{15}{32}$	$\frac{3}{8}$	$\frac{7}{8}$	$\frac{15}{16}$	$7\frac{3}{4}$	$3\frac{7}{8}$	$15\frac{1}{4}$	$9\frac{3}{4}$	$1\frac{7}{16}$	2	$2\frac{3}{4}$	87.9
$\frac{1}{2}$	$\frac{7}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	8	4	$15\frac{1}{4}$	$9\frac{3}{4}$	$1\frac{7}{16}$	2	$2\frac{3}{4}$	88.3
$\frac{17}{32}$	$\frac{7}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	8	4	$15\frac{1}{4}$	$9\frac{3}{4}$	$1\frac{7}{16}$	2	$2\frac{3}{4}$	88.3
$\frac{9}{16}$	$\frac{7}{16}$	1	$1\frac{1}{16}$	8	4	17	11	$1\frac{5}{8}$	$2\frac{1}{4}$	3	86.7
$\frac{19}{32}$	$\frac{1}{2}$	1	$1\frac{1}{16}$	8	4	17	11	$1\frac{5}{8}$	$2\frac{1}{4}$	3	86.7
$\frac{5}{8}$	$\frac{1}{2}$	1	$1\frac{1}{16}$	8	4	17	11	$1\frac{5}{8}$	$2\frac{1}{4}$	3	86.7
$\frac{21}{32}$	$\frac{1}{2}$	1	$1\frac{1}{16}$	8	4	17	11	$1\frac{5}{8}$	$2\frac{1}{4}$	3	86.7
$\frac{11}{16}$	$\frac{1}{2}$	$1\frac{1}{8}$	$1\frac{3}{16}$	$8\frac{1}{4}$	$4\frac{1}{8}$	$18\frac{1}{2}$	12	$1\frac{13}{16}$	$2\frac{3}{8}$	$3\frac{1}{4}$	85.6
$\frac{23}{32}$	$\frac{1}{2}$	$1\frac{1}{8}$	$1\frac{3}{16}$	$8\frac{1}{4}$	$4\frac{1}{8}$	$18\frac{1}{2}$	12	$1\frac{13}{16}$	$2\frac{3}{8}$	$3\frac{1}{4}$	85.6
$\frac{3}{4}$	$\frac{1}{2}$	$1\frac{1}{8}$	$1\frac{3}{16}$	$8\frac{1}{4}$	$4\frac{1}{8}$	$18\frac{1}{2}$	12	$1\frac{13}{16}$	$2\frac{3}{8}$	$3\frac{1}{4}$	85.5
$\frac{25}{32}$	$\frac{9}{16}$	$1\frac{1}{4}$	$1\frac{5}{16}$	$8\frac{1}{2}$	$4\frac{1}{4}$	$20\frac{1}{4}$	$13\frac{1}{4}$	2	$2\frac{5}{8}$	$3\frac{1}{2}$	84.6
$\frac{13}{16}$	$\frac{9}{16}$	$1\frac{1}{4}$	$1\frac{5}{16}$	$8\frac{1}{2}$	$4\frac{1}{4}$	$20\frac{1}{4}$	$13\frac{1}{4}$	2	$2\frac{5}{8}$	$3\frac{1}{2}$	84.6
$\frac{27}{32}$	$\frac{9}{16}$	$1\frac{1}{4}$	$1\frac{5}{16}$	$8\frac{1}{2}$	$4\frac{1}{4}$	$20\frac{1}{4}$	$13\frac{1}{4}$	2	$2\frac{5}{8}$	$3\frac{1}{2}$	84.2
$\frac{7}{8}$	$\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{5}{16}$	$8\frac{3}{4}$	$4\frac{3}{8}$	$20\frac{1}{4}$	$13\frac{1}{4}$	2	$2\frac{5}{8}$	$3\frac{1}{2}$	84.1
$\frac{29}{32}$	$\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{5}{16}$	$8\frac{3}{4}$	$4\frac{3}{8}$	$20\frac{1}{4}$	$13\frac{1}{4}$	2	$2\frac{5}{8}$	$3\frac{1}{2}$	83.6
$\frac{15}{16}$	$\frac{11}{16}$	$1\frac{1}{4}$	$1\frac{5}{16}$	9	$4\frac{1}{2}$	$20\frac{1}{4}$	$13\frac{1}{4}$	2	$2\frac{5}{8}$	$3\frac{1}{2}$	83.7
$\frac{31}{32}$	$\frac{11}{16}$	$1\frac{1}{4}$	$1\frac{5}{16}$	9	$4\frac{1}{2}$	$20\frac{1}{4}$	$13\frac{1}{4}$	2	$2\frac{5}{8}$	$3\frac{1}{2}$	83.2
1	$\frac{3}{4}$	$1\frac{3}{8}$	$1\frac{7}{16}$	$9\frac{1}{2}$	$4\frac{3}{4}$	22	$14\frac{1}{2}$	$2\frac{3}{16}$	2 [illegible]/8	$3\frac{3}{4}$	83.4

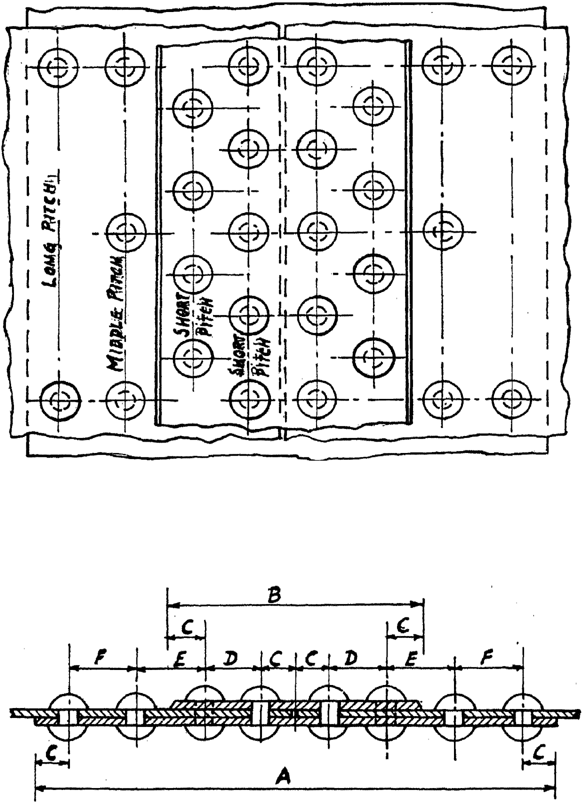


Fig. 44. Butt Joint--Quadruple Riveted
(Recommended by Hartford S.B.I. & I. Co.)

Table 22. - Butt Joints--Quadruple Riveted (Fig. 44)
(Recommended by Hartford S.B.I.& I. Co.)

Thickness of Plate, Inches	Thickness of Straps, Inches	Diam. of Cold Rivet, Inches	Diam. of Rivet Hole, Inches	Long Pitch, Inches	Middle Pitch, Inches	Short Pitch, Inches	A Inches	B Inches	C Inches	D Inches	E Inches	F Inches	Efficiency Per Cent
1/4	1/4	5/8	11/16	11	5 1/2	2 3/4	16	7 3/4	1 1/16	1 3/4	2 1/8	2 1/4	93.8
9/32	1/4	5/8	11/16	11	5 1/2	2 3/4	16 1/2	7 3/4	1 1/16	1 3/4	2 1/8	2 1/4	93.8
5/16	9/32	3/4	13/16	13	6 1/2	3 1/4	18 7/8	8 3/4	1 1/4	1 7/8	2 7/16	2 5/8	93.8
11/32	9/32	3/4	13/16	13	6 1/2	3 1/4	18 7/8	8 3/4	1 1/4	1 7/8	2 7/16	2 5/8	93.8
3/8	5/16	3/4	13/16	14	7	3 1/2	19 1/8	8 3/4	1 1/4	1 7/8	2 7/16	2 3/4	94.2
13/32	5/16	3/4	13/16	14	7	3 1/2	19 1/8	8 3/4	1 1/4	1 7/8	2 7/16	2 3/4	94.2
7/16	3/8	7/8	15/16	15 1/2	7 3/4	3 7/8	21 3/8	9 3/4	1 7/16	2	2 3/4	3 1/16	94.0
15/32	3/8	7/8	15/16	15 1/2	7 3/4	3 7/8	21 3/8	9 3/4	1 7/16	2	2 3/4	3 1/16	94.0
1/2	7/16	7/8	15/16	16	8	4	21 1/2	9 3/4	1 7/16	2	2 3/4	3 1/8	94.1
17/32	7/16	7/8	15/16	16	8	4	21 1/2	9 3/4	1 7/16	2	2 3/4	3 1/8	94.1
9/16	7/16	1	1 1/16	16	8	4	23 5/8	11	1 5/8	2 1/4	3	3 5/16	93.4
19/32	1/2	1	1 1/16	16	8	4	23 5/8	11	1 5/8	2 1/4	3	3 5/16	93.4
5/8	1/2	1	1 1/16	16	8	4	23 5/8	11	1 5/8	2 1/4	3	3 5/16	93.4
21/32	1/2	1	1 1/16	16	8	4	23 5/8	11	1 5/8	2 1/4	3	3 5/16	93.4
11/16	1/2	1 1/8	1 3/16	16 1/2	8 1/4	4 1/8	25 5/8	12	1 13/16	2 3/8	3 1/4	3 9/16	92.8
23/32	1/2	1 1/8	1 3/16	16 1/2	8 1/4	4 1/8	25 5/8	12	1 13/16	2 3/8	3 1/4	3 9/16	92.8
3/4	1/2	1 1/8	1 3/16	16 1/2	8 1/4	4 1/8	25 5/8	12	1 13/16	2 3/8	3 1/4	3 9/16	92.7
25/32	9/16	1 1/4	1 5/16	17	8 1/2	4 1/4	27 7/8	13 1/4	2	2 5/8	3 1/2	3 13/16	92.3
13/16	9/16	1 1/4	1 5/16	17	8 1/2	4 1/4	27 7/8	13 1/4	2	2 5/8	3 1/2	3 13/16	92.3
27/32	9/16	1 1/4	1 5/16	17	8 1/2	4 1/4	27 7/8	13 1/4	2	2 5/8	3 1/2	3 13/16	91.8
7/8	5/8	1 1/4	1 5/16	17 1/2	8 3/4	4 3/8	28	13 1/4	2	2 5/8	3 1/2	3 7/8	91.2
29/32	5/8	1 1/4	1 5/16	17 1/2	8 3/4	4 3/8	28	13 1/4	2	2 5/8	3 1/2	3 7/8	90.5
15/16	11/16	1 1/4	1 5/16	18	9	4 1/2	28 1/8	13 1/4	2	2 5/8	3 1/2	3 15/16	90.1
31/32	11/16	1 1/4	1 5/16	18	9	4 1/2	28 1/8	13 1/4	2	2 5/8	3 1/2	3 15/16	89.5
1	3/4	1 3/8	1 7/16	19	9 1/2	4 3/4	30 1/2	14 1/2	2 3/16	2 7/8	3 3/4	4 1/4	90.2
1 1/32	3/4	1 3/8	1 7/16	19	9 1/2	4 3/4	30 1/2	14 1/2	2 3/16	2 7/8	3 3/4	4 1/4	89.6
1 1/16	3/4	1 3/8	1 7/16	19	9 1/2	4 3/4	30 1/2	14 1/2	2 3/16	2 7/8	3 3/4	4 1/4	89.0
1 3/32	3/4	1 3/8	1 7/16	19	9 1/2	4 3/4	30 1/2	14 1/2	2 3/16	2 7/8	3 3/4	4 1/4	88.5
1 1/8	3/4	1 3/8	1 7/16	19	9 1/2	4 3/4	30 1/2	14 1/2	2 3/16	2 7/8	3 3/4	4 1/4	88.0
1 5/32	3/4	1 3/8	1 7/16	19	9 1/2	4 3/4	30 1/2	14 1/2	2 3/16	2 7/8	3 3/4	4 1/4	87.5
1 3/16	13/16	1 3/8	1 7/16	20	10	5	30 5/8	14 1/2	2 3/16	2 7/8	3 3/4	4 5/16	87.7
1 7/32	13/16	1 3/8	1 7/16	20	10	5	30 5/8	14 1/2	2 3/16	2 7/8	3 3/4	4 5/16	87.2
1 1/4	7/8	1 3/8	1 7/16	20	10	5	30 5/8	14 1/2	2 3/16	2 7/8	3 3/4	4 5/16	86.8

Table 23. - Lap-Riveted Joints for Girth Seams of Cylindrical Vessels with Solid Unstayed Heads
(Recommended by The Hartford S.B.I. & I. Co.)

Thickness of Plate, Inches	Diameter of Rivet Holes, Inches	Diameter of Cold Rivet, Inches	Single-Riveted Lap Joints		Double-Riveted Lap Joints	
			Pitch of Rivets, Inches	Efficiency of Joint, Per Cent	Pitch of Rivets, Per Cent	Efficiency of Joint, Per Cent
$\frac{1}{4}$	$\frac{11}{16}$	$\frac{5}{8}$	$1\frac{3}{4}$	60.7		
$\frac{9}{32}$	$\frac{11}{16}$	$\frac{5}{8}$	$1\frac{3}{4}$	60.3		
$\frac{5}{16}$	$\frac{13}{16}$	$\frac{3}{4}$	$1\frac{7}{8}$	56.7		
$\frac{11}{32}$	$\frac{13}{16}$	$\frac{3}{4}$	$1\frac{7}{8}$	56.7		
$\frac{3}{8}$	$\frac{13}{16}$	$\frac{3}{4}$	$1\frac{7}{8}$	56.7		
$\frac{13}{32}$	$\frac{13}{16}$	$\frac{3}{4}$	$1\frac{7}{8}$	54.5		
$\frac{7}{16}$	$\frac{15}{16}$	$\frac{3}{4}$	$2\frac{1}{8}$	55.9		
$\frac{15}{32}$	$\frac{15}{16}$	$\frac{3}{4}$	$2\frac{1}{8}$	55.4		
$\frac{1}{2}$	$\frac{15}{16}$	$\frac{7}{8}$	$2\frac{1}{8}$	52.0		
$\frac{17}{32}$	$\frac{15}{16}$	$\frac{7}{8}$	$2\frac{1}{16}$	50.4	3	68.8
$\frac{9}{16}$	$1\frac{1}{16}$	1	$2\frac{3}{8}$	53.3	$3\frac{1}{4}$	67.3
$\frac{19}{32}$	$1\frac{1}{16}$	1			$3\frac{1}{4}$	67.3
$\frac{5}{8}$	$1\frac{1}{16}$	1			$3\frac{1}{4}$	67.3
$\frac{21}{32}$	$1\frac{1}{16}$	1			$3\frac{1}{4}$	66.5
$\frac{11}{16}$	$1\frac{3}{16}$	$1\frac{1}{8}$			$3\frac{3}{4}$	68.3
$\frac{23}{32}$	$1\frac{3}{16}$	$1\frac{1}{8}$			$3\frac{3}{4}$	65.8
$\frac{3}{4}$	$1\frac{3}{16}$	$1\frac{1}{8}$			$3\frac{3}{4}$	63.0
$\frac{25}{32}$	$1\frac{5}{16}$	$1\frac{1}{4}$			4	67.2
$\frac{13}{16}$	$1\frac{5}{16}$	$1\frac{1}{4}$			4	66.6
$\frac{27}{32}$	$1\frac{5}{16}$	$1\frac{1}{4}$			4	64.1
$\frac{7}{8}$	$1\frac{5}{16}$	$1\frac{1}{4}$			4	61.9
$\frac{29}{32}$	$1\frac{5}{16}$	$1\frac{1}{4}$			4	59.7
$\frac{15}{16}$	$1\frac{5}{16}$	$1\frac{1}{4}$			4	57.7
$\frac{31}{32}$	$1\frac{5}{16}$	$1\frac{1}{4}$			4	55.9
1	$1\frac{7}{16}$	$1\frac{3}{8}$			$4\frac{1}{2}$	57.7
$1\frac{1}{32}$	$1\frac{7}{16}$	$1\frac{3}{8}$			$4\frac{1}{2}$	56.0
$1\frac{1}{16}$	$1\frac{7}{16}$	$1\frac{3}{8}$			$4\frac{1}{2}$	54.3
$1\frac{3}{32}$	$1\frac{7}{16}$	$1\frac{3}{8}$			$4\frac{1}{2}$	52.8
$1\frac{1}{8}$	$1\frac{7}{16}$	$1\frac{3}{8}$			$4\frac{1}{2}$	51.3
$1\frac{5}{32}$	$1\frac{7}{16}$	$1\frac{3}{8}$			$4\frac{1}{2}$	49.9
$1\frac{3}{16}$	$1\frac{7}{16}$	$1\frac{3}{8}$			$4\frac{1}{2}$	48.6
$1\frac{7}{32}$	$1\frac{7}{16}$	$1\frac{3}{8}$			$4\frac{1}{2}$	47.4
$1\frac{1}{4}$	$1\frac{7}{16}$	$1\frac{3}{8}$			$4\frac{1}{2}$	46.2

In tank and boiler work it is customary to use the same size of rivets in the girth seams as in the longitudinal seams. The strength of the girth joint has to be at least 50% of that required for the longitudinal joint.

Chapter XV

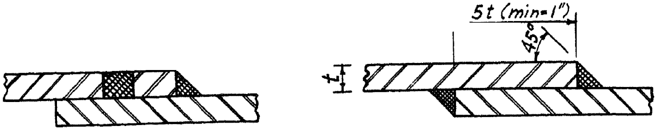
FUSION-WELDED JOINTS

Types of Joint. - Thin sheets, about #20 G. and less, are flared, and then the edge is welded. Heavier sheets and plates are either lap-welded or butt-welded.

In a lap joint the edges of the plates overlap and are joined by a fillet weld, which is a weld of triangular cross section joining two surfaces approximately at right angles to each other. Fillet welds which are applied to join two plates have to be built up to the full thickness of the plate and are called full-fillet welds, while fillet welds which are applied in order to tighten a seam are called seal welds. If a fillet weld is placed at the edge of each of two overlapping plates the joint is called a Double Full-Fillet Lap Joint. If only one edge is fillet-welded the joint is a Single Full-Fillet Lap Joint. Lap joints have a low efficiency on account of the bending stresses produced in them, and should not be used where great strength is required.

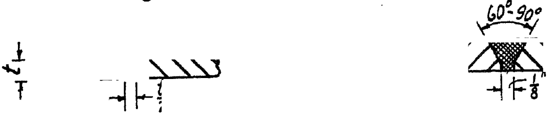
A Butt-Welded Joint is formed by the fusion of two abutting edges with weld metal. For thicknesses larger than 1/4 in., welding grooves are provided, which may be either V- or U-shaped in cross section. Plates up to 1/2 in. thick are Single-Welded, i.e., the weld metal is added from one side only. For plates larger than 1/2 in. thickness a Double-Welded Butt Joint should be used which is produced by adding weld metal from both sides of the joint. The different types of joints are shown in Fig. 45. The angle of V-shaped grooves formed by the beveled edge of the two plates is usually made approximately 70 degrees.

A universal joint which is applicable to all thicknesses and requires only two different bevelling tools, is shown in Fig. 46. The welding bevels for pipe and fittings, as specified by the ASA Pipe Standards, 1939, are given in Fig. 47.



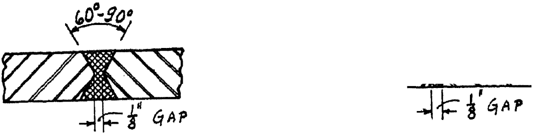
a) Single Full-Fillet Lap Joint with Plug Welds

b) Double Full-Fillet Lap Joint



c) Single-Welded Butt Joint with Square Edge Plates 1/4 in. and Less

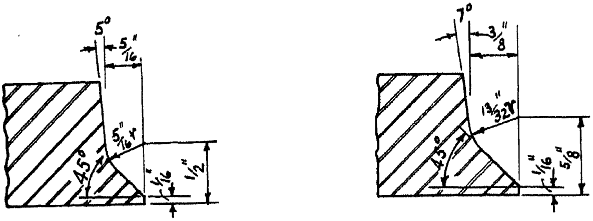
d) Single-Welded Butt Joint with V-Type Groove Plates 1/4 to 1/2 in. Thick



e) Double-Welded Butt Joint with V-Type Groove Plates Larger than 1/2 in.

f) Double-Welded Butt Joint with U-Type Groove Plates Larger than 1/2 in.

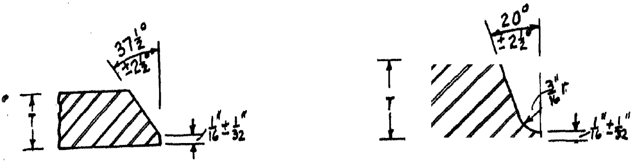
Fig. 45. Types of Welded Joints



a) Longitudinal Weld All Thicknesses

b) Circumferential Weld All Thicknesses

Fig. 46. Universal Welding Joint



a) Standard Straight Bevel of Welding Ends for Thicknesses (T) 3/16 in. to 3/4 in., Inclusive

b) Standard U-Bevel of Welding Ends for Thickness (T) Greater than 3/4 in.--This form is not recommended for Acetylene Welding and may be changed to straight bevel.

Fig. 47. Welding Bevels for Pipe and Fitting (A.S.A. Pipe Standards, 1939)

Efficiency of Welded Joints. - The efficiency of welded joints varies with the grade of steel, type of joint, and the quality of work. The efficiency values assigned to the different types of joints are usually meant to be applied to average workmanship. If the main welded joints of a vessel are examined by radiograph and the defects repaired, credit is given for the higher degree of workmanship and inspection and the average efficiency values may be increased by multiplying with a factor of 1.12. For vessels which are stress-relieved the average efficiency value may be multiplied by 1.06.

The A.P.I.- A.S.M.E. Code gives the following tabulation of the maximum efficiencies to be used for the various types of joints made up of firebox grade steel and before stress relieving and radiographing:

Type of Joint	Limitations	Efficiency of Joint %
Double-welded butt joint	None	80
Single-welded butt joint, with backing-up strip	Joints not over 1¼" thick	80

Type of Joint	Limitations	Efficiency of Joint %
Single-welded butt joint, without backing-up strip	Joints not over 5/8 in. thick	70
Double full-fillet lap joint	Circumferential joints only, not over 5/8 in. thick	65
Single full-fillet lap joints with plug welds	Circumferential joints only, not over 5/8 in. thick	65
Single full-fillet lap joints without plugs	Only for attaching heads convex to pressure and plates not over 5/8 in. thick	55

Allowance for workmanship:

Radiographing	1.12
Stress relieving	1.06

Other factors to be considered in figuring the efficiency of a seam are

Shear factor	80% of joint efficiency
Side weld factor	75% of shear stress
Circular welds around nozzles and openings	87.5% of shear stress

The application of these factors may be illustrated by two examples:

a) Circular fillet weld attaching welding nozzle to shell, stress relieved only.
Overall efficiency = joint efficiency X circular weld factor X shear factor X stress relieving = .65 X .875 X .8 X 1.06 = 48.3%

b) Vertical fillet weld attaching support lug to shell, not X-rayed or stress-relieved.

Overall efficiency = Joint efficiency
X side weld factor
X shear factor
.65 X .75 X .8 = 39%

For carbon steel, below 650 deg. F., the allowable stress of the weld (b) would equal 13,750 X .39 = 5,360 lb. per sq. in.

For fillet welds, the throat dimension of the fillet is used in stress calculations, which for a bevel of 45 degrees equals 0.707 times the side of the fillet. The allowable load of a 3/8 in. fillet weld, for instance, would be .375 X .707 X 5,360 = 1,420 lb. per lineal inch.

REFERENCES

1. Welding Handbook, 1938, American Welding Society.

Chapter XVI

COPPER VESSELS

Tensile Strength. - The tensile strength of copper begins to decrease at 250°F and for temperatures exceeding 500°F copper should not be used at all. Since the strength of copper also depends on the mechanical treatment, it is presumed that all copper parts of vessels are properly hardened by cold working. For this purpose the vessels have to be so designed that they can be hammered all over. For temperatures up to 250°F a tensile strength of 30,000 lbs. per square inch can be used. The factor of safety is usually 4.5 to 5. Maximum allowable working stresses for higher temperatures, as specified by the A.S.M.E. Code for Unfired Pressure Vessel Code, are given in Table A-3.

Wall Thickness. - For the calculation of wall thicknesses, in general the same formulas apply as for steel vessels. Where the effect of the cold working is diminished through a successive treatment, f.i. tinning of the hammered part, the thickness should be increased about 0.02 to 0.04 inches, the larger value applying to thin plates. Allowance has to be made also for corrosion and for a reduction of the thickness due to the working of the metal. Par. U-38 of the A.S.M.E. Code specifies that the reduction of the plate thickness due to the forming operation should not exceed 10% of the thickness computed by the formula.

External Pressure. - Copper cylinders under external pressure may be calculated by the equations given for steel vessels. However, since the modulus of elasticity E of copper is constant only at stresses below the proportional limit and decreases rapidly as the stress is increased above this limit, the thickness and collapsing pressure can be obtained only by a trial and error method. This will be seen from the following consideration. The shell thickness is expressed as a function of the modulus of elasticity in the design formulas. On the other hand, the modulus varies with the stress which in turn depends on the thickness of the shell. It is, therefore, necessary to assume a value of E and figure the shell thickness. Having thus obtained the shell thickness, the working stress in the shell can be calculated by the formula:

$$S = \frac{p\ D}{2\ t} \tag{1}$$

Then the value of the reduced modulus E' is determined for this stress by means of Timoshenko's formula for the reduced modulus (Ref. 3):

$$E' \quad \frac{4\ E\ E_1}{(\sqrt{E} + \sqrt{E_1})^2} \tag{2}$$

In this formula E is the constant value of the modulus below the proportional limit and E_1 is the slope of the stress-strain curve at the stress for which the modulus is to be determined. If the reduced modulus of elasticity thus computed does not check the modulus originally assumed, another value of the reduced modulus has to be assumed and the calculation to be repeated until a sufficient agreement is obtained. For values of E' see Fig. 183 (p. 277).

The yield points of copper sheet below 250°F are: soft 10,000, cold rolled 12-15,000, hard 34,000 lb. per sq. in. In all cases, the tensile modulus below the proportional limit is $E = 15 \times 10^6$ lb. per sq. in.

Approximate values of the t/D ratio at normal temperatures have been charted in Fig. 184, p. 278, which is based on a yield point $S_y = 10{,}000$, Poisson's ratio $\mu = 0.33$, and a constant modulus $E = 15{,}000{,}000$. In the construction of the chart, the following formulas from Chap. VII were applied: eq. (9) for the left section, eq. (4a) for the middle section, and eq. (5) for the horizonal section of the curves at the right.

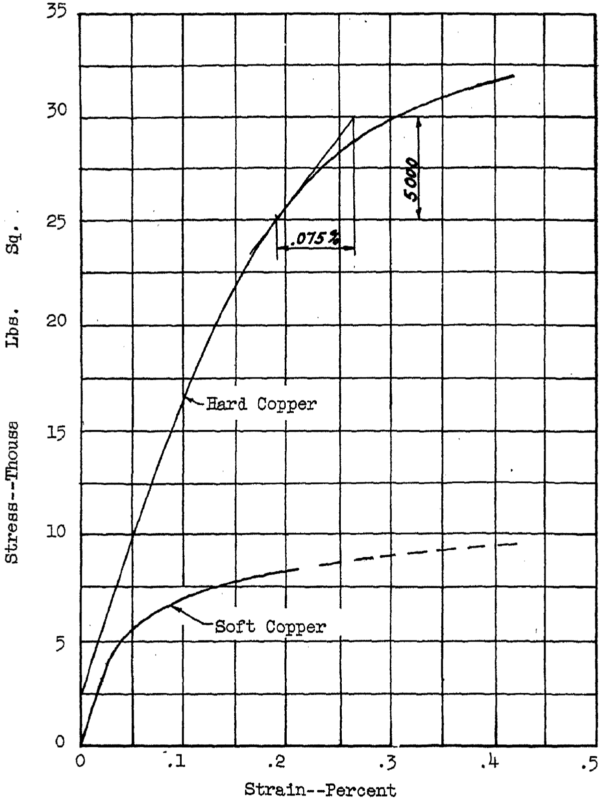


Fig. 48. Stress--Strain Diagram for Copper

Example. - Determine the reduced modulus E' of hard copper at a stress of 25,000 psi.

Solution: From Fig. 48, obtain $E_1=5000/0.00075=$ 6,666,666 psi. Insert E=15,000,000 psi and E_1 in equation (2) and calculate E'=9,600,000 psi.

Dished Heads. - The computation of spherical bottoms and dished heads is the same as for steel.

Spherical bottoms are either flanged, as shown in Fig. 49, and bolted together, or joined by riveting, as shown in Fig. 50. The free rim a is stiffened by an iron rod over which the metal is bent. In Fig. 51 and 52 a flat bar and an angle iron, respectively, are used to stiffen the free edge.

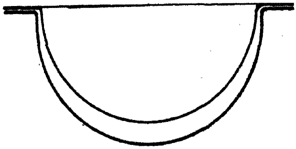


Fig. 49

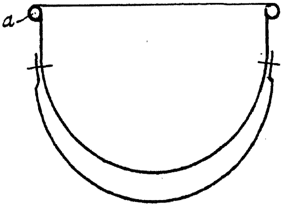


Fig. 50

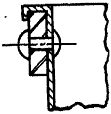


Fig. 51

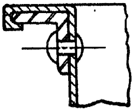


Fig. 52

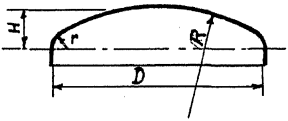


Fig. 53

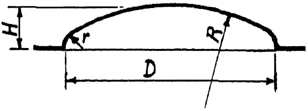


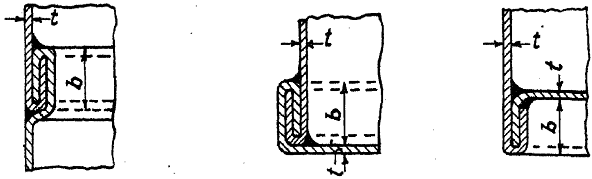
Fig. 54

The dished heads, shown in Fig. 53 and Fig. 54, usually have the following proportions:

for small pressure:
$H = 0.1\ D$ and $R = 1.3\ D$

for higher pressure:
$H = 0.134\ D$ and $R = D$

JOINING OF COPPER SHEETS. Lock Seams. - Sheets up to 0.083 in. in thickness (#14 Stubs Gage) for low pressures (principally fluid pressure) are joined by lock seams. Fig. 55a shows a double lock seam joint in a cylindrical shell. Several ways of lock seaming a bottom and a cylindrical shell are illustrated in Fig. 55.



(a) (b) (c)

Fig. 55

Lock seams are not tight and have to be soft soldered in order to produce perfect tightness. If the seams are not accessible from the inside, they may be carried out as shown in Fig. 56. Most hard rolled sheets of the non-ferrous metals are not malleable enough for this operation and should be ordered with a suitable temper. The edges of cold rolled copper are usually annealed before they are bent.

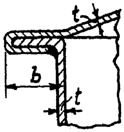


Fig. 56

The width b of the seam varies with the thickness t of the sheet. Usual dimensions are:

t =	.02535	.03196	.0403	.04526	.05082	.06408	.07196	.08081	in.
	22	20	18	17	16	14	13	12	B. & S. Ga.
b =	$\frac{5}{16}$"	$\frac{3}{8}$"	$\frac{7}{16}$"	$\frac{1}{2}$"	$\frac{9}{16}$"	$\frac{5}{8}$"	$\frac{11}{16}$"	$\frac{3}{4}$"	

The longitudinal seam should be brazed or welded.

Riveted Joints. - Riveting is used to attach fittings and fixtures to copper work, and for seaming and joining operations, particularly in cases where brazing is not permissible on account of the annealing effect. It allows the use of cold rolled sheets and thus eliminates the necessity of hammering. The rivets and sheets are usually ring-set and sweated so that a perfectly smooth finish is produced on the inside. Sweating is necessary to make the seam leak-proof. Sheets of 1/4 in. thickness or over can be caulked.

Riveted and Sweated Joints. - Fig. 57 shows a typical single-riveted and ring-set joint and Fig. 58 a double-riveted joint. Typical dimensions for both joints are given in Table 24.

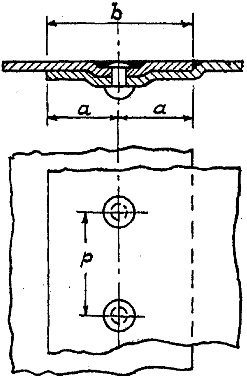


Fig. 57

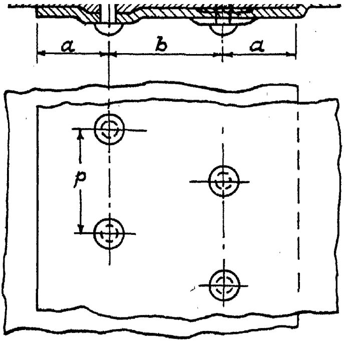


Fig. 58

In Fig. 59 a copper jacket is attached to a hemi-sphericall copper bottom by a double-riveted joint. In Fig. 60 a bottom is riveted to a shell. The corner radius r is made about 1 to 2 inches.

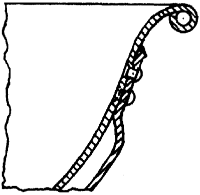


Fig. 59

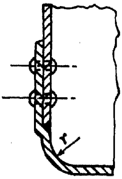


Fig. 60

Table 24. - Dimensions of Riveted and Sweated Copper Seams as per Fig. 57 and Fig. 58

Thickness of Sheet t inches	Rivet		Single Riveted		Double Riveted			
	Diameter d inches	No.	Pitch P inches	Lap b = 2a inches	Pitch P inches	Margin a inches	Back Pitch b inches	Lap c inches
$\frac{1}{16}$	$\frac{11}{64}$	00	$1\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{7}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$2\frac{3}{8}$
$\frac{3}{32}$	$\frac{3}{16}$	0	$1\frac{1}{2}$	$1\frac{1}{2}$	2	$\frac{3}{4}$	$\frac{7}{8}$	$2\frac{3}{8}$
$\frac{1}{8}$	$\frac{1}{4}$	1	$1\frac{5}{8}$	$1\frac{3}{4}$	$2\frac{1}{8}$	$\frac{7}{8}$	1	$2\frac{3}{4}$
$\frac{5}{32}$	$\frac{9}{32}$	2	$1\frac{3}{4}$	$1\frac{3}{4}$	$2\frac{1}{4}$	$\frac{7}{8}$	1	$2\frac{3}{4}$
$\frac{3}{16}$	$\frac{19}{64}$	2	$1\frac{7}{8}$	2	$2\frac{3}{8}$	$\frac{7}{8}$	$1\frac{1}{8}$	$2\frac{7}{8}$
$\frac{7}{32}$	$\frac{11}{32}$	4	2	2	$2\frac{1}{2}$	$\frac{7}{8}$	$1\frac{1}{8}$	$2\frac{7}{8}$

For higher pressures the longitudinal seam is usually double-riveted while the circumferential seam is single-riveted.

With thin sheets and where a smooth surface is required, oval head braziers' rivets are used which have a flat head and are more suitable for ring-setting. One usual type is shown in Fig. 61. Braziers' rivets are specified by their number. Dimensions and weights of standard sizes are given in Table 25.

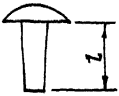


Fig. 61

Riveted and Caulked Joints. - Copper plates, about 1/4" thick and above, can be caulked. For this type of seam, round or cone head copper rivets are taken, dimensions of which are shown in Table 26, on following page. The weights of round head copper rivets are given in Table 27, page 152.

Table 25. - Oval Head Copper Brazier's Rivets as per Fig. 61

Number	00	0	1	2	3	4	5	6	7	8	9	10
Diam. of Head, in.	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{23}{32}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{15}{16}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$
Diam. of Shank Under Head, in.	$\frac{11}{64}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{19}{64}$	$\frac{11}{32}$	$\frac{23}{64}$	$\frac{25}{64}$	$\frac{7}{16}$	$\frac{17}{32}$	$\frac{5}{8}$	$\frac{21}{32}$
ℓ, in.	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{15}{16}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{4}$
Number to lb.	224	138	65	45	38	25	20	17	13	7	5	$4\frac{1}{2}$

Table 26.--Dimensions of Round Head Copper Rivets

Diameter of Head, in.	Thickness of Head, in.	Diameter of Shank, in.
$\frac{7}{32}$	$\frac{3}{32}$	$\frac{1}{8}$
$\frac{21}{64}$	$\frac{9}{64}$	$\frac{3}{16}$
$\frac{7}{16}$	$\frac{3}{16}$	$\frac{1}{4}$
$\frac{35}{64}$	$\frac{15}{64}$	$\frac{5}{16}$
$\frac{21}{32}$	$\frac{9}{32}$	$\frac{3}{8}$

Can be furnished from 1/32 in. diameter to 3/4 in. diameter.

Table 27. - Approximate Number of Pieces Per Pound of Round Head Copper Rivets

Length in inches	Diameter of Shank--Inches											
	$\frac{1}{16}$	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$
$\frac{1}{8}$	4,500	..	..	..	..	..	..	..	..	..	..	..
$\frac{3}{16}$	3,700	1,250	640	470	..	..	..	..	..	..	..	..
$\frac{1}{4}$	3,000	1,100	585	400	225	90	..	..	..	..	..	..
$\frac{3}{8}$	2,200	870	448	340	180	78	46	..	..	..	..	..
$\frac{1}{2}$	1,750	700	372	280	144	74	42	27	..	..	..	..
$\frac{3}{4}$	1,250	520	271	205	112	57	33	22	..	..	..	..
$\frac{7}{8}$	1,050	470	230	180	100	50	30	20	..	..	..	..
1	950	410	205	165	90	45	28	18	11	$8\frac{3}{4}$	$5\frac{1}{4}$	$3\frac{1}{2}$
$1\frac{1}{4}$	..	..	150	135	75	38	24	16	10	$7\frac{3}{4}$	$4\frac{3}{4}$	$3\frac{1}{4}$
$1\frac{1}{2}$	..	..	130	120	64	34	21	14	9	7	$4\frac{1}{4}$	3
$1\frac{3}{4}$	..	..	120	105	57	30	19	12	$8\frac{1}{2}$	6	$3\frac{3}{4}$	$2\frac{2}{3}$
2	..	..	..	92	50	27	17	11	8	$5\frac{3}{4}$	$3\frac{1}{2}$	$2\frac{1}{2}$
$2\frac{1}{4}$	..	..	..	..	44	25	15	10	7	$5\frac{1}{4}$	$3\frac{1}{3}$	$2\frac{1}{4}$
$2\frac{1}{2}$	..	..	..	..	40	23	13	9	$6\frac{1}{2}$	5	3	2
$2\frac{3}{4}$	..	..	..	..	..	21	11	$8\frac{1}{2}$	6	$4\frac{1}{2}$	$2\frac{3}{4}$	$1\frac{7}{8}$
3	..	..	..	..	..	19	10	8	$5\frac{1}{2}$	$4\frac{1}{4}$	$2\frac{2}{3}$	$1\frac{3}{4}$
$3\frac{1}{4}$	..	..	..	..	..	..	..	$7\frac{1}{2}$	$5\frac{1}{4}$	4	$2\frac{1}{2}$	$1\frac{2}{3}$
$3\frac{1}{2}$	..	..	..	..	..	..	..	7	5	$3\frac{3}{4}$	$2\frac{1}{3}$	$1\frac{5}{8}$
$3\frac{3}{4}$	..	..	..	..	..	..	..	$6\frac{1}{2}$	$4\frac{3}{4}$	$3\frac{1}{2}$	$2\frac{1}{4}$	$1\frac{1}{2}$
4	..	..	..	..	..	..	..	6	$4\frac{1}{2}$	$3\frac{1}{4}$	2	$1\frac{3}{8}$

Table 28.--Ratio of Head Diameters to Shank Diameters

Style of Head	Round or Button Head	Oval Head	Flat Head	Counter-sunk Head--80°	Cone Head
Dia. of Head	1.75 x dia. of shank	1.87 x dia. of shank	2 x dia. of shank	2 x dia. of shank	1.75 x dia. of shank
Height of Head	0.75 x dia. of shank	0.6 x dia. of shank	0.33 x dia of shank	0.6 x dia. of shank	0.87 x dia. of shank

A single-riveted and caulked lap-joint for copper sheets is shown in Fig. 62, and a double-riveted joint in Fig. 63. On account of the softness of the metal, straps are used for higher pressure, as illustrated in Fig. 64.

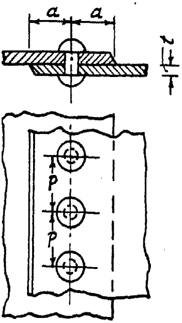


Fig. 62

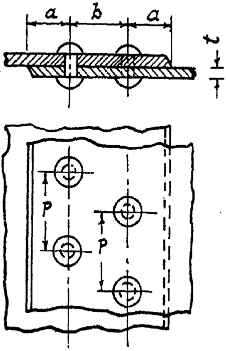


Fig. 63

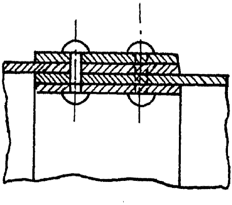


Fig. 64

Bottoms are attached to shells according to Fig. 65 and Fig. 66, on following page.

The design of a riveted and caulked joint for copper is similar to the one for iron. Since copper is softer than iron and offers less resistance to caulking it is a good practice to select a smaller pitch and rivets of a somewhat smaller diameter than for iron.

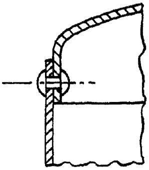


Fig. 65

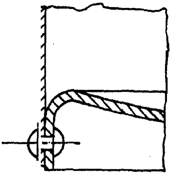


Fig. 66

Approximately the following proportions may be used for a copper sheet of thickness t, all dimensions in inches:

(a) Single riveted and caulked copper joint
diameter of rivet $d = 2\ t$
pitch $p = 2.5$ to $3\ d$
marginal distance $a = 1\frac{1}{2}$ to $2\ d$

(b) Double riveted and caulked copper joint
diameter of rivet $d = 2\ t$
pitch $p = 3\ d + \frac{1}{4}$ in.
marginal distance $a = 1.5$ to $2\ d$
back pitch $b = 0.5\ p$

Efficiency of Riveted Joints. - For the calculation of the joint efficiency the following ultimate stresses may be used up to 250°F

(a) Copper plate:

Tensile strength	30,000 lbs. per sq. in.
Crushing strength	40,000 lbs. per sq. in.

(b) Copper rivets
Ultimate strength of rivets in shear:

in single shear	21,000 lbs. per sq. in.
in double shear	42,000 lbs. per sq. in.

In accordance with these data, dimensions of single and double riveted joints have been compiled in Tables 29 and 30, for plates from $\frac{1}{4}$" to $\frac{1}{2}$" thick.

Table 29. - Dimensions of Copper Lap Joint, Single Riveted, as per Fig. 62, in Copper Plate, inches

Thickness of Plate t	Diameter of Rivet d	Diameter of Hole D	Pitch P	Marginal Distance a	Efficiency of Joint %
$\frac{1}{4}$	$\frac{1}{2}$	$\frac{9}{16}$	$1\frac{1}{2}$	$\frac{7}{8}$	46.5
$\frac{9}{32}$	$\frac{1}{2}$	$\frac{9}{16}$	$1\frac{1}{2}$	$\frac{7}{8}$	41.5
$\frac{5}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$1\frac{7}{8}$	1	43.8
$\frac{11}{32}$	$\frac{5}{8}$	$\frac{11}{16}$	$1\frac{7}{8}$	1	40.3
$\frac{3}{8}$	$\frac{3}{4}$	$\frac{13}{16}$	$2\frac{1}{4}$	$1\frac{1}{4}$	43.2
$\frac{13}{32}$	$\frac{3}{4}$	$\frac{13}{16}$	$2\frac{1}{4}$	$1\frac{1}{4}$	39.5
$\frac{7}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$2\frac{1}{4}$	$1\frac{1}{4}$	37.0
$\frac{7}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	$2\frac{5}{8}$	$1\frac{1}{2}$	42.1

Table 30. - Dimensions of Copper Lap Joint, Double Riveted, as per Fig. 63, in Copper Plate, inches

Thickness of Plate t	Diameter of Rivet d	Diameter of Hole D	Pitch P	Marginal Distance a	Back Pitch b	Efficiency of Joint %
$\frac{1}{4}$	$\frac{1}{2}$	$\frac{9}{16}$	$1\frac{7}{8}$	$\frac{7}{8}$	$1\frac{1}{4}$	70.0
$\frac{9}{32}$	$\frac{1}{2}$	$\frac{9}{16}$	$1\frac{7}{8}$	$\frac{7}{8}$	$1\frac{1}{4}$	66.0
$\frac{5}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$2\frac{1}{4}$	1	$1\frac{1}{2}$	69.8
$\frac{11}{32}$	$\frac{5}{8}$	$\frac{11}{16}$	$2\frac{1}{4}$	1	$1\frac{1}{2}$	67.3
$\frac{3}{8}$	$\frac{3}{4}$	$\frac{13}{16}$	$2\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{3}{4}$	69.0
$\frac{13}{32}$	$\frac{3}{4}$	$\frac{13}{16}$	$2\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{3}{4}$	67.8
$\frac{7}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$2\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{3}{4}$	63.0
$\frac{7}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	3	$1\frac{1}{2}$	$1\frac{7}{8}$	69.0
$\frac{15}{32}$	$\frac{3}{4}$	$\frac{13}{16}$	$2\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{3}{4}$	58.8
$\frac{15}{32}$	$\frac{7}{8}$	$\frac{15}{16}$	3	$1\frac{1}{2}$	$1\frac{7}{8}$	68.7
$\frac{1}{2}$	$\frac{3}{4}$	$\frac{13}{16}$	$2\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{3}{4}$	55.0
$\frac{1}{2}$	$\frac{7}{8}$	$\frac{15}{16}$	3	$1\frac{1}{2}$	$1\frac{7}{8}$	65.6

EXAMPLE:

(a) Single riveted lap joint, copper
$S_t = 30{,}000$, $S_s = 21{,}000$, $S_c = 40{,}000$,
plate thickness $t = 5/16$ in., pitch $P = 1\text{-}7/8$ in., diameter of rivet shank $d = 5/8$ in., diameter of rivet hole $D = 11/16$ in.

Failure of joint through:

(1) Shearing of rivets $= \frac{\pi D^2}{4} \times S_s = .371 \times 21{,}000$

$= 7{,}800$ lb.

(2) Tearing of plate between rivets

$= (P - D)t\ S_t$
$= (1.875 - .6875) \times .3125 \times 30{,}000$
$= 11{,}150$ lb.

(3) Crushing of rivet or plate

$= D \times t \times S_c$
$= .6875 \times .3125 \times 40{,}000$
$= 8{,}620$ lb.

Strength of solid plate

$= P \times t \times S_t$
$= 1.875 \times .3125 \times 30{,}000$
$= 17{,}800$ lb.

Joint strength = 7,800 lb. (least of values (1), (2), (3))

Joint efficiency $= \frac{7{,}800}{17{,}800} = 43.8\%$

(b) Double riveted lap joint, copper
plate thickness $t = 5/16$ in., pitch $P = 2\frac{1}{4}$ in., diameter of rivet shank $d = 5/8$ in., diameter of rivet hole $D = 11/16$ in.

Failure of joint through:

(1) Shearing of rivets $= \frac{2\ D^2}{4} \times S = 2 \times .371$
$\times\ 21{,}000 = 15{,}600$ lb.

(2) Tearing of plate between rivets

$= (P - D)t \times S$

$= (2.25 - .6875) \times .3125 \times 30{,}000$

$= 14{,}700$ lb.

(3) Crushing of rivet or plate

$= 2DtS_c = 2 \times .6875 \times .3125 \times 40{,}000$

$= 17{,}240$ lb.

Strength of solid plate

$= PtS_t = 2.25 \times .3125 \times 30{,}000$

$= 21{,}100$ lb.

Joint strength = 14,700 lb.

$$\text{Joint efficiency} = \frac{14{,}700}{21{,}100} = 69.8\%$$

Soldering, Brazing, Welding

Soft Solder. - This joint is made by lapping the edges and soft soldering them and is applicable only for low temperatures and low stresses as it is lacking in strength. It is principally used to join tubing and for leak-proofing lock seams and riveted seams.

Soft solders are composed of tin and lead in various proportions. About even parts of lead and tin are used for soldering and tinning, while for wiping a mixture of at least two parts of lead to one part of tin is taken. The melting point increases with the content of lead as follows:

Lead	55	40	35	%
Tin	45	60	65	%
Melting Point, approx.	320	275	280	°F

The strength of a soldered joint depends on the composition of the solder and on the size of the area. The higher the content of tin the stronger is the solder. With increasing lap the unit strength of the joint decreases because it becomes more difficult to obtain a perfect distribution and binding of the solder. One square inch of a good soldered joint can stand approximately 360 lbs. pull. With reference to the sheet it is figured that the strength of the solder is equal to the strength of the copper sheet if the soldered area is 13 times as large as the cross section of the copper. In order to take care of defective soldering the width of the lap is made 20 x thickness of copper sheets and tubes.

Hard Soldering or Brazing. - Plates up to about 1/2 in. thickness can be joined by hard soldering. In this operation an actual transfusion of the molten metal takes place and a joint is produced which has practically the same strength as the plate. Hard solders for copper work consist of copper, zinc and tin. For nickel work silver is added. Various compositions and their melting points are given in Table 31. For brazing copper plates the zinc content is usually 50 per cent.

The two edges are chamfered or thin-edged so that the thickness of the metal will be about even when the two parts are joined (see Fig. 67). Since the plates warp when they are heated, they have to be held together. Tinned tack rivets which are sometimes used are objectionable insofar as they tend to leak. A better method is to "dove-tail" the seam. For this purpose a number of "clamps" are cut with a chisel into one edge, and the other edge is inserted therein. This method has the disadvantage that sometimes cracks develop at the end of the cuts. The sheets are usually annealed.

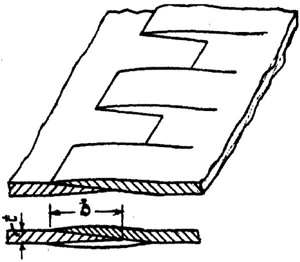


Fig. 67

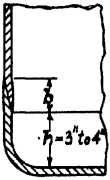


Fig. 68

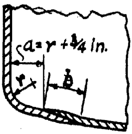


Fig. 69

Table 31.--Hard Brass Solders
(American Brass Company)

Brazing Solder Number	Grain	Size	Color	Melting Point °C	°F
40	Round	Extra Fine	Yellow	882	1620
41	"	Fine	"	"	"
42	"	Med. Fine	"	"	"
43	"	Med. Coarse	"	"	"
44	"	Coarse	"	"	"
45	"	Extra Coarse	"	"	"
51	"	Fine	"	"	"
52	"	Med. Fine	"	"	"
61	"	Fine	Gray	813	1495
62	"	Med. Fine	"	"	"
91	"	Fine	"	868	1595
92	"	Med. Fine	"	"	"
100	Long	Extra Fine	Yellow	882	1620
101	"	Fine	"	"	"
103	"	Med. Fine	"	"	"
105	"	Coarse	"	"	"
106	"	Extra Coarse	"	"	"
500	Lump		"	"	"
520	"		"	"	"
1200	Long	Coarse	"	"	"
1407	Round	Fine to Med. Coarse Mixed	"	"	"
Black Button	"	Fine	Black	782	1440

The lap b is made approximately 5 x thickness of plates as follows:

t = $\frac{1}{16}$ $\frac{1}{8}$ $\frac{3}{16}$ $\frac{1}{4}$ $\frac{5}{16}$ $\frac{3}{8}$ $\frac{7}{16}$ $\frac{1}{2}$ inches

b = $\frac{3}{8}$ $\frac{5}{8}$ $\frac{15}{16}$ 1 $1\frac{1}{4}$ $1\frac{3}{8}$ $1\frac{1}{2}$ $1\frac{1}{2}$ inches

When a head is joined to a cylindrical shell the brazed seam may be located in the shell according Fig. 68 or in the head according Fig. 69.

In contact with certain chemicals the zinc content is eaten out of the solder through electrolytic action thus causing a porous seam. This may be prevented by adding lead or silver or by the use of silver solder.

Silver Solder. - Silver solder is used where corrosive action does not permit the use of brass solder or where the greater fluidity and penetrative power of silver is of advantage. Various kinds of silver solders and their melting points are given in Table 32.

Table 32.--Melting Temperatures and Colors of "Handy" Silver Solders
(Handy and Harman)

Name	Melting Point °F	Color	Name	Melting Point °F	Color
T L	1600	Brass Yellow	D E	1370	Yellow White
A T T	1500		E T	1340	
N E	1575		E T X	1425	
N T	1450	Pale Yellow	R T	1325	Silver White
S S	1435		B T	1425	
D T	1445		I T	1460	

Sil-Fos is the trade-name of a patented hard solder which contains a small percentage of silver and is produced by Handy and Harman. It is not as corrosion resistant as the silver solders, but is free flowing and lower in cost. The melting point is about 1,300°F.

Welding of Copper. - The welding of copper offers certain difficulties due to its high heat conductivity and due to the fact that it forms Cu_2O very easily in the melted state which makes the weld porous and brittle and reduces the strength of the weld quite considerably. To counteract this tendency, various deoxidizing substances are utilized of which silicon and phosphorus are the most prominent ones. Up to 4%

silicon is alloyed with copper directly to produce the so-called deoxidized copper. It increases the strength and ductility of copper and does not affect its malleability. Phosphorus is applied with the welding rod in the form of phosphor bronze welding rods, which contain about 96 per cent copper, 3.75 per cent tin, and 0.25 phosphorus. This weld is stronger but less malleable than a joint welded with base metal filler. Ordinary commercial copper is not satisfactory for welding operations.

Of the various welding methods gas welding with the oxy-acetylene torch is generally preferred. As far as electric welding is concerned, welding rods for metallic arc welding have not been sufficiently perfected as yet, while fairly good results have been obtained with carbon arc welding. Considerable progress has been made in metallic arc welding by the application of the shielded arc process.

The preparation of the edge of copper sheets for welding varies with the thickness, as shown in Fig. 70.

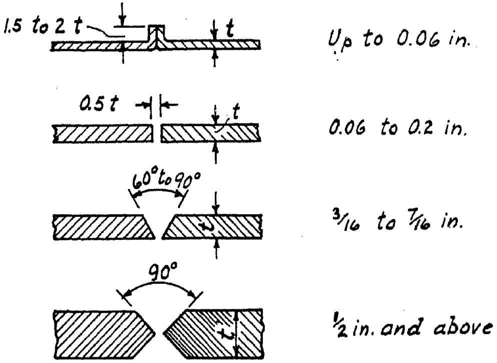


Fig. 70. Preparation of Copper Sheets for Welding

The Research Laboratories of the Union Carbide and Carbon Company subjected welded copper plates to tension tests the results of which are presented in Table 33.

Table 33.--Tension Tests of Welded Copper Seams
(Union Carbide & Carbon Res. Labor.)

Material	Breaking Strength, lb. per sq. in.	Elongation in 2 in., per cent	Remarks
Average copper	32,000	45	
Welds in copper containing 0.01 to 0.08 per cent oxygen (commercial copper)	15,500	8	Breaks a small fraction of an inch from the weld.
Welds in deoxidixed copper	26,250	23	Break in weld.

Tinning. - Copper kettles are very often tinned to protect the copper against corrosive action or vice versa to protect the substance handled against contamination by the copper. In most cases a composition of one part tin and two parts lead is used, with the exception of containers for foodstuffs which are tinned with pure block tin.

Flanged Joints. - Flanged joints made up of brass flanges or angle rings riveted or soldered to the shell are expensive and rarely used, except for small diameters and manholes. In most cases the copper is flanged over an iron backing ring. The gasket surface usually extends over the entire width of the iron ring, as in Fig. 71. Joints where the copper flange reaches to the inside of bolts only, as in Fig. 72, are cheaper but not suitable for higher pressure because of the bending stresses set up in the iron flanges by the gasket pressure. Sometimes angle rings are used according Fig. 73.

The inner corner of the flanges should have as large a radius as possible.

Covers are bolted on to shells as shown in Fig. 74. If greater stiffness is desired, as f.i. for

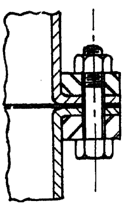


Fig. 71

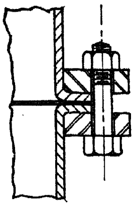


Fig. 72

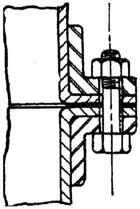


Fig. 73

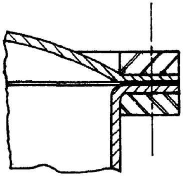


Fig. 74

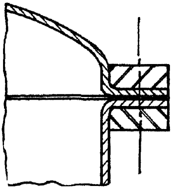


Fig. 75

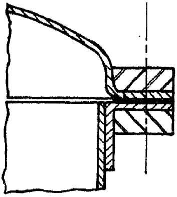


Fig. 76

vacuum vessels, the head is dished as in Fig. 75. Since the flanging operation weakens the shell, sometimes a flanged copper ring, which can be made of heavier metal, is slipped over and soldered or riveted to the shell, as in Fig. 76, to supply additional strength.

Jacketed copper bottoms are bolted on to the shell according Fig. 77. If a steel jacket is used instead of copper, the joint may be designed as shown in Fig. 78. The ring "a" in Fig. 79 serves to reduce the bending moment produced by the bolts.

The size of the ring flange depends on the thickness of the shell and the size of the bolts. It has to be sufficiently rigid so that it is not deformed when the copper shell is flanged over it and when the joint is tightened. The dimensions given in Table 34 for loose steel flanges of copper expansion joints may be used as a guidance.

The number of bolts is usually divisible by four. For small vessels which are supported by three legs a number divisible by three is selected. In order to have

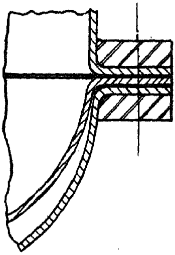


Fig. 77

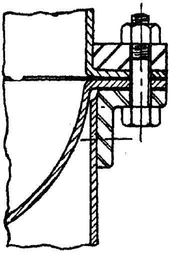


Fig. 78

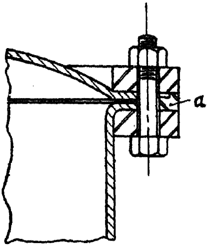


Fig. 79

sufficient clearance for the wrench, the bolt circle is made from 1/8 to 1/4 in. larger than the mean diameter of the iron ring. If an angle ring is used as backing-up ring the bolt circle is still further increased in order to provide sufficient room for the nut. The calculation of the bolting is the same as for steel vessels.

For quick-opening joints very often swing bolts are used. For small diameters and manholes the rings are bronze castings with a tongue and groove joint. They are either riveted to the shell, as in Fig. 80, or hard soldered according Fig. 81.

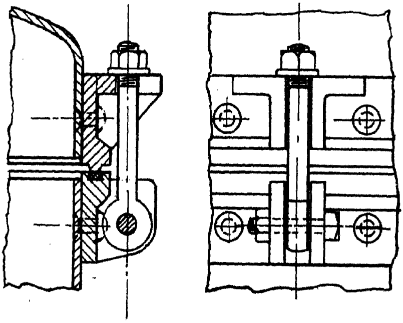


Fig. 80

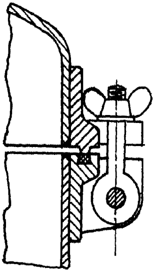
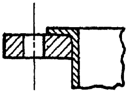
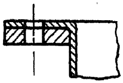


Fig. 81

Table 34. - Dimensions of Loose Steel Flanges for Copper Expansion Joints--10 to 25 lbs. per sq. in. Pressure or Vacuum



Type A



Type B

Nominal Pipe Size Inches	Outs. Dia. of Flange Inches	Cast Iron Inches	Steel Type A Inches	Steel Type B Inches	Bolt Circle Inches	Number of Bolts	Bolt Size Inches
6	11	1	1	$\frac{3}{8}$	$9\frac{1}{2}$	8	$\frac{3}{4}$
8	$13\frac{1}{2}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$\frac{1}{2}$	$11\frac{3}{4}$	8	$\frac{3}{4}$
10	16	$1\frac{3}{16}$	$1\frac{1}{8}$	$\frac{1}{2}$	$14\frac{1}{4}$	12	$\frac{7}{8}$
12	19	$1\frac{1}{4}$	$1\frac{1}{4}$	$\frac{1}{2}$	17	12	$\frac{7}{8}$
14	21	$1\frac{3}{8}$	$1\frac{3}{8}$	$\frac{1}{2}$	$18\frac{3}{4}$	12	1
16	$23\frac{1}{2}$	$1\frac{7}{16}$	$1\frac{3}{8}$	$\frac{1}{2}$	$21\frac{1}{4}$	16	1
18	25	$1\frac{9}{16}$	$1\frac{1}{2}$	$\frac{5}{8}$	$22\frac{3}{4}$	16	$1\frac{1}{8}$
20	$27\frac{1}{2}$	$1\frac{11}{16}$	$1\frac{1}{2}$	$\frac{5}{8}$	25	20	$\frac{7}{8}$
24	32	$1\frac{7}{8}$	$1\frac{1}{2}$	$\frac{3}{4}$	$29\frac{1}{2}$	20	$\frac{7}{8}$
30	$38\frac{3}{4}$	$2\frac{1}{8}$	$1\frac{3}{4}$	$\frac{7}{8}$	36	28	1
36	46	$2\frac{3}{8}$	$1\frac{3}{4}$	1	$42\frac{3}{4}$	32	1
42	53	$2\frac{5}{8}$	$1\frac{7}{8}$	$1\frac{1}{8}$	$49\frac{1}{2}$	36	$1\frac{1}{8}$
48	$59\frac{1}{2}$	$2\frac{3}{4}$	$1\frac{7}{8}$	$1\frac{1}{4}$	56	44	$1\frac{1}{8}$
54	$66\frac{1}{4}$	3	$2\frac{1}{8}$	$1\frac{3}{8}$	$62\frac{3}{4}$	44	$1\frac{1}{8}$
60	73	$3\frac{1}{8}$	$2\frac{1}{8}$	$1\frac{1}{2}$	$69\frac{1}{4}$	52	$1\frac{1}{4}$
72	$86\frac{1}{2}$	$3\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{3}{4}$	$82\frac{1}{2}$	60	$1\frac{1}{4}$

Bolt holes are drilled $\frac{1}{8}$" larger than the bolt diameters. Flanges 20 in. and larger are shown drilled to the 25-pound low pressure standard.

Connections in Copper Vessels. - Small screwed connections are provided by means of various kinds of bushings. For soft soldering the bushings carry wide flanges, while for brazing a small chamfered flange is provided and a groove which allows setting of the copper in preparation for brazing (Fig. 82D). Brazing fittings are made of a composition containing about 80% copper and 20% zinc and have a wall thickness of about five times the depth of the thread. A flanged socket, which is used for soft soldering, is shown in Fig. 82A. Bosses are brazed into the shell according Fig. 82B and 82C. A collar of sufficient height is raised out of the copper to facilitate brazing. In Fig. 82C the boss is set at an oblique angle and, therefore, has to be slanted at one end to obtain a smooth surface.

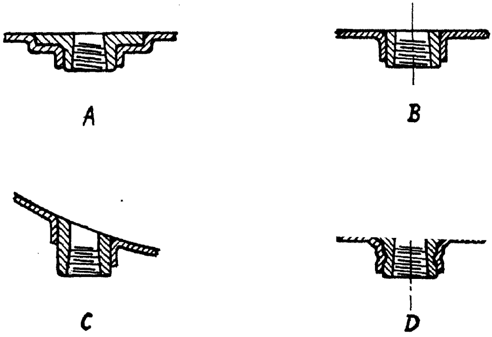


Fig. 82

Drains of jacketed kettles are fashioned along similar lines. They are equipped with a ring, which acts as spacer between the two bottoms, a washer which is usually beveled to fit the outer bottom, and a lock nut. The thread is cut straight to allow tightening of the lock nut, with the exception of the tapered part at the end. A soldering drain is shown in Fig. 83 and a brazing drain in Fig. 84.

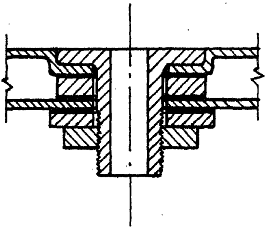


Fig. 83

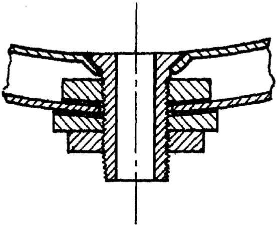


Fig. 84

Larger drains are cast with a step which takes the place of the spacer ring, as shown in Fig. 85.

Flanged connections are made either by means of brazed or soldered pipe connections or by inserting a brass fitting of which there are numerous different designs suiting particular cases. A typical Van Stone type joint is shown in Fig. 86. Large connections are worked out of the metal of the vessel itself according Fig. 87.

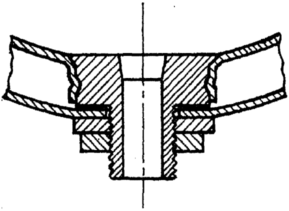


Fig. 85

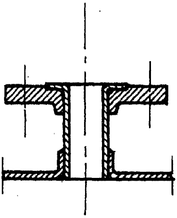


Fig. 86

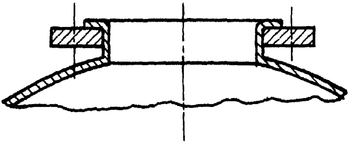


Fig. 87

In Fig. 88 a drain pipe is welded to a collar raised out of the bottom of a tank. A sweated and riveted drain is shown in Fig. 89.

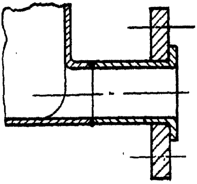


Fig. 88

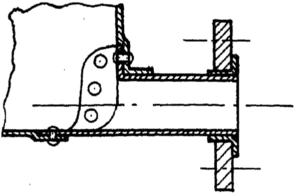


Fig. 89

Brass and bronze nozzles are bolted on to the shell, as in Fig. 90, or riveted as in Fig. 91.

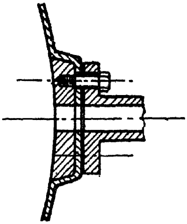


Fig. 90

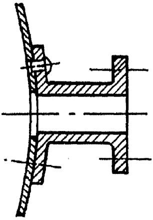


Fig. 91

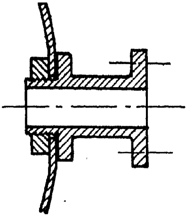


Fig. 92

The brass plate in Fig. 90 is either brazed or riveted and sweated. The nozzle in Fig. 92 is threaded on one end and fastened by a lock nut from the inside of the vessel. Fig. 93 shows how a pipe is brought through the wall of a vessel.

A stuffing box for an agitator shaft is illustrated in Fig. 94. There are several ways of reinforcing the opening in the head. In this case a brass pad has been riveted onto the head and the stuffing box has been bolted against the pad with stud bolts.

Copper Pipe Joints. - Permanent joints are made by means of a cup joint, Fig. 95, or a sleeve joint, Fig. 96. The end of the tube, respectively of the sleeve, is

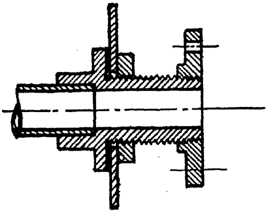


Fig. 93

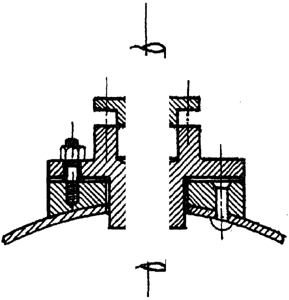


Fig. 94

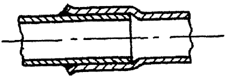


Fig. 95

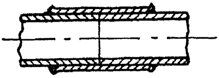


Fig. 96

flared a little and brazed. Up to 6 in. diameter the depth of the cup is approximately equal to the diameter.

For screwed joints, brass unions are brazed onto the pipe. There are a number of different types on the market, of which one very common type is shown in Fig. 97.

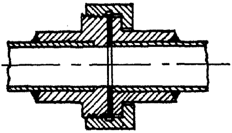


Fig. 97

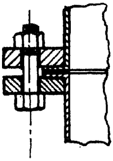


Fig. 98

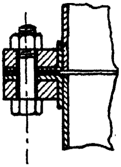


Fig. 99

Flanged joints may be made in several different ways. Fig. 98 shows a Van Stone joint. The flanges are usually cast iron and recessed. Instead of flanging the end of the tube, collars are sometimes brazed on the tube as shown in Fig. 99.

Brazed brass flanges, Fig. 100, are preferred where a greater strength is required. The copper content of the brass composition should be at least 75%. Steel flanges can be brazed to copper pipes quite satisfactorily, while cast-iron flanges are less suitable for this purpose.

For pipes, 30 in. diameter and larger, high collared flanges or angle rings of brass or iron are riveted and sweated on, as shown in Fig. 101. The same type of joint is used down to 10 in. diameter if the wall thickness is less than #14 Stubs Gage or 0.083 in.

In Fig. 102 a lens joint is shown which is used for high pressure work.

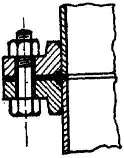


Fig. 100

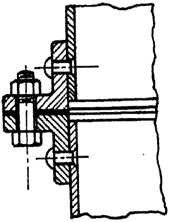


Fig. 101

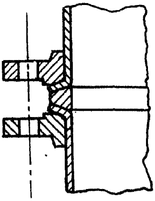


Fig. 102

Layout of Copper Bottoms and Domes. - In fabricating small copper kettles, consisting of a cylindrical shell and a semi-spherical bottom, first a cylindrical shell is formed and partly drawn in at the bottom, as shown in Fig. 103. Then a copper disc of proper diameter is bumped to the same semi-spherical shape and attached to the drawn-in end of the shell. The diameter of the disc usually equals approximately one-third of the semi-circle.

Ordinarily semi-spherical bottoms below 24" diameter are formed from one sheet. The diameter of this circle is made approximately 5 per cent less than the arc of the bottom, because the metal is drawn out while the sheet is worked. Another empirical rule is to make the diameter of the circle equal to the diameter of the flange plus the depth of the dish.

Larger bottoms are formed from a disc and a cone

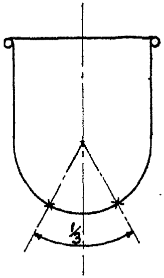


Fig. 103

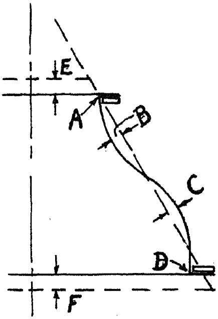


Fig. 105

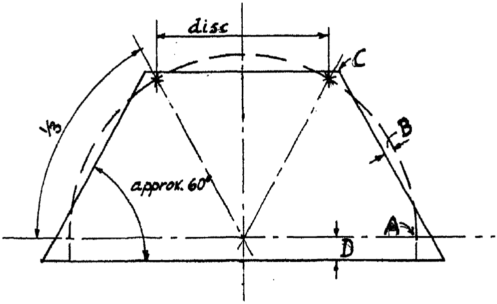


Fig. 104

frustum, as shown in Fig. 104. The disc equals approximately 1/3 of the arc. The frustum is laid out so that the amount of bumping out and drawing in of metal is nearly equalized. This means that, in Fig. 104, the cone should have the same distance from the semi-circle at A, B, and C. The distance D represents an allowance to be made for the lap or flange.

Dome shaped covers, Fig. 105, are also formed out of a frustum of a cone. The layout is based on the same principles as hemispherical bottoms, i.e., the work of bumping out and drawing in metal should be about equal. The distances of the cone to the semi-circle at

B, C, and D should, therefore, be nearly equal. E and F are allowances for the upper and lower flange. The exact size of the plate required for the frustum is then obtained by developing the cone in the customary manner.

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2. G. Hoennicke, Handbuch zum Dampffass--und Apparatebau. Julius Springer, Berlin, 1924.
3. S. Timoshenko, Strength of Materials.

Chapter XVII

EVERDUR VESSELS

Design. - Everdur is equal to steel in tensile strength and the design of Everdur vessels is in general treated very much like that of steel vessels.

The strength values of annealed Everdur (1010) tank plates are:

Tensile strength, min.	50,000 lb. per sq. in.
Yield point, min.	18,000 lb. per sq. in.
Elongation in 2 inches	40 per cent
Modulus of Elasticity	15,000,000 lb. per sq. in.

Maximum allowable working stresses are the same as the stresses given for copper-silicon alloy plates in Table U-4 revised of the A.S.M.E. Code, which is 10,000 lbs. per sq. in. up to 250°F, and 5,000 lbs. per sq. in. up to 350°F.

Welding. - Everdur can easily be welded by the oxy-acetylene torch and by the various electric welding methods, such as metallic and carbon arc, electric resistance seam or spot welding, using Everdur rods. It is successfully welded to steel and copper by the carbon or metallic arc. Everdur welding rod is used in butt welding galvanized iron by means of the carbon arc.

Everdur sheets 3/16 in. thick and less are butt welded without beveling the edges. The method most generally used for this thickness is carbon arc welding, using a grooved back-up bar of copper and clamps to keep the sheet in alignment. Joints set up as shown in Fig. 106 may be welded by the carbon arc without a welding rod after a suitable arc welding flux has been applied. Carbon arc welds of upstanding edges may be made in a similar manner without the addition of any metal from a welding rod. Various types of lap joints employed in joining the heads and shells of light gage storage tanks

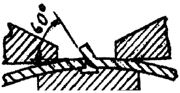


Fig. 106. Set Up for Carbon Arc Welding Without Filler Rod. Sheets 3/16 in. thick and less.

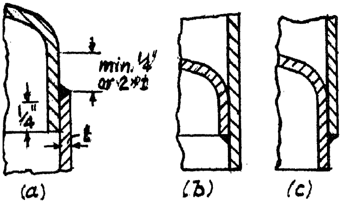


Fig. 107. Head and Shell Joints. Sheets 3/16 in. thick and less.

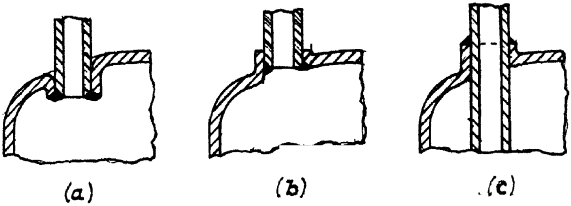


Fig. 108

are shown in Fig. 107. Tubular connections in these heads may be made by one of the methods shown in Fig. 108.

Plates over 3/16" thick are usually butt welded after the edges have been beveled. Types and degree of bevel as recommended by the American Brass Company are given in Table 35. The double V butt weld should be used wherever it is possible to weld from both sides of the plates.

Riveting. - Due to the ease with which Everdur can be welded, riveting is hardly ever resorted to in pressure vessel work. However, the following ultimate unit values are suggested for the design of riveted joints:

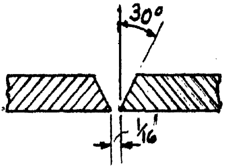
(a) Everdur tank plate

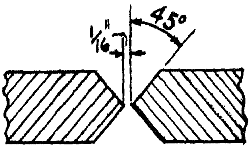
Tensile strength	50,000 lbs. per sq. in.
Crushing strength	80,000 " " " "

(b) Everdur (1015) rivets (cold headed)
Ultimate strength in shear
in single shear 40,000 lbs. per sq. in.
in double shear 80,000 " " " "

Table 35. Butt Joints for Everdur Plates

Metal Thickness Inches	Electric Arc Welding		Oxy-Acetylene Welding	
	Type of Bevel	Degree of Bevel	Type of Bevel	Degree of Bevel
¼	double V	45	single V	45
	single V	45		
⅜	double V	45	double V	45
	single V	30	single V	30
½	double V	45	double V	30
	single V	30	single V	30
⅝	double V	45	double V	30
	single V	30	single V	30
¾	double V	30	double V	30
	single V	30	single V	30





REFERENCE

1. Bulletins published by The American Brass Company, Waterbury, Connecticut.

Chapter XVIII

ALUMINUM VESSELS

Design Formulas. - For the computation of the wall thickness of cylindrical shells and dished heads the same formulae apply as for iron. Additions should be made to the theoretical values, particularly for thin sheets, where rigidity is the important factor or where the thickness is decreased through forming.

In the design of vessels subjected to external pressure the same method has to be applied as for copper vessels, since the modulus of elasticity of aluminum also varies with the stress.

Design Stresses. - The material adjacent to welds is usually in the annealed condition. Therefore, the design of welded aluminum vessels is generally based on the strength of the material in the annealed condition, even if a harder temper is used, as is often the case with cylindrical shells, or if most of the material has been hardened through working after the forming operation.

The factor of safety allowed is 4 for low pressure and 5 for high pressure and certain hazardous types of equipment. For storage tanks under atmospheric pressure stresses approaching the yield strength may be used.

Mechanical properties based on data published by the Aluminum Company of America are given in Table 8, and factors for design stresses at elevated temperatures in Table 7.

It may be noted, however, that a tendency exists to use more and more the "0.2% elastic limit" instead of the ultimate tensile strength as the basis of design for light metals. As will be seen from the stress-strain diagram, Fig. 109, aluminum does not have a precisely defined elastic limit, yield point, and limit of proportionality. It, therefore, is customary to assume as the elastic limit a stress which leaves a

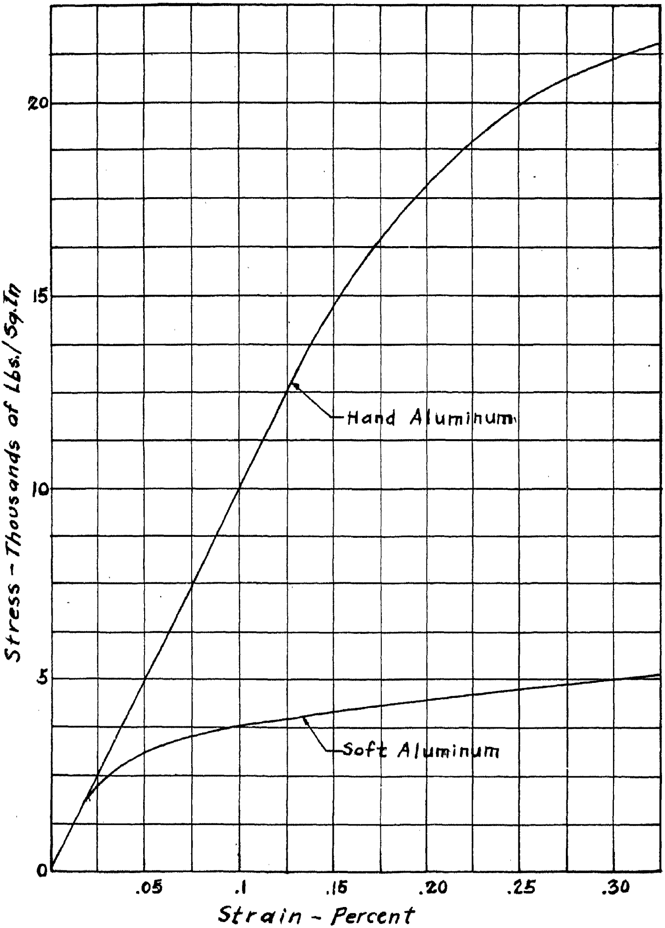


Fig. 109. Stress-Strain Diagram of Aluminum.

permanent set of 0.02% and is accordingly called the "0.02% elastic limit." Similarly, the "0.02% elastic limit" has been substituted for the yield point.

Design of Vessels. - The design of aluminum vessels follows in general the same lines as for copper vessels, except that the welding, which is very extensively used with aluminum, requires a different treatment of some construction details. Dished heads have about the same proportions as the copper heads, in Figs. 53 and 54. Some typical examples of aluminum work are the steam jacketed kettle in Fig. 110 and the flanged kettle in Fig. 111.

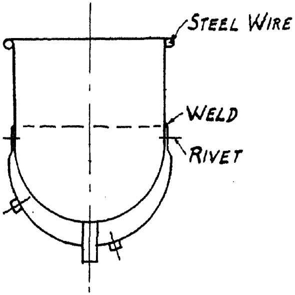


Fig. 110

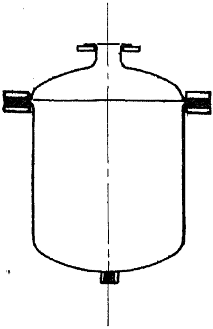


Fig. 111

Riveted Joints. - Aluminum plates can be joined by riveting in a satisfactory manner. The aluminum alloys most commonly used for rivets are: 2S, 3S, 4S, 17S and 51S. Steel rivets are sometimes used for aluminum structures, if the joint is painted afterwards.

Rivets of 2S, 3S and 4S aluminum are furnished in an intermediate temper and are driven cold in the "as-received" condition. Rivets of 17S and 51S in sizes up to 1/2 to 3/4 in. are driven cold; above 3/4 in. they are driven hot. They should be heat treated before and during the driving operation in order to improve their strength. The heat treatment consists of heating the rivets to 930°F to 950°F and quenching them. During the driving operation the cold tools and the metal around the hole produce a certain quenching affect.

Hard rivets should not be driven in soft plates, but it is sometimes of advantage to use a soft rivet for a somewhat harder plate. In general, it is a good practice to keep rivet and plate of the same properties. Recommended combinations of plate and rivets are given in Table 36.

Table 36. Combinations of Plates and Rivets of Various Properties

Plate	Rivet
2 S, any temper	2 S
3 S-0, 4 S-0	3 S
3 S or 4 S, quarter hard or harder	3 S, 4 S, 51 S-W
A 17 S-T, 17 S-T, 27 S-T	17 S, Steel
51 S-W, 51 S-T, 53 S-W, 53 S-T	4 S, 51 S-W, Steel

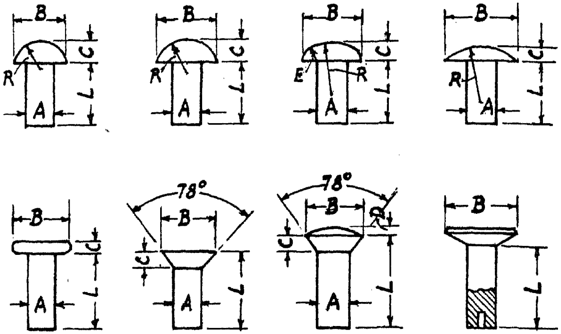
Types of Rivets. - Aluminum alloy rivets are furnished in several different types, general dimensions of which are given in Table 37. Button, round, mushroom, brazier, and flat head rivets have a small fillet at the junction of the shank and head the radius of which is equal to about one-tenth of the shank diameter with a minimum of 0.01 inches.

Length of Rivets. - The length of the rivet depends on the grip, the clearance between hole and rivet, the alloy, and the form of the head. Values for a number of different sizes and grips under the stated conditions are given in Table 38. Since conditions vary it is always good to determine the length by driving a few test rivets.

Rivet Holes. - Rivet holes in aluminum alloys are either punched, drilled, or sub-punched and reamed to size with bridge reamers of the spiral-fluted type. The diameter of the hole should be about 1/16 in. larger than the diameter of the rivet.

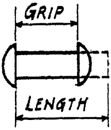
Table 37. Common Types of Aluminum Rivets

(By Aluminum Company of America)



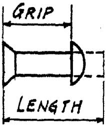
Kind	Width B	Head Depth C	Head Radius R	Edge Radius E	Oval Depth D
Button Head	1.75 A	0.75 A	0.885 A	......	.....
Round Head	2 A	0.75 A	1.042 A	......	.....
Mushroom Head	2 A	0.625 A	1.634 A	0.50 A	.....
Brazier Head	2.50 A	0.50 A	1.8125 A	......	.....
Flat Head	2 A	0.4 A	....	......	.....
Ctsk. Head	1.81 A	0.5 A	....	......	.....
Ctsk. Oval Head	1.81 A	0.5 A	1.7656 A	......	0.25
Tubular Shank	This rivet made with several sizes of heads.				

Design of Riveted Joints. - The calculation of riveted aluminum joints is based on the same principals as given for iron rivets. Values of safe shearing stresses for aluminum rivets are given in Table 39. These values are based on a rather low factor of safety because tests have proven that in some cases the driving operation increases the shearing strength. In Table 39 the diameter of the rivet is assumed to be equal to the diameter of the hole, which is limited to a value 5 per cent greater than the nominal diameter of the rivet. Some engineers, however, base their calculations on the nominal diameter of the rivet.



Case 1

Table 38. - Length of Rivets for Various Grips
17 S rivets in 17 S-T plates, button heads
Values above heavy line apply to Case 1 only.
(By Aluminum Company of America)



Case 2

Grip	Diameter Rivet								
	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1
	Hole diameter $\frac{1}{32}$ inch greater than rivet diameter								
$\frac{1}{8}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	...	...	...	...	...	...
$\frac{1}{4}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{5}{16}$	$1\frac{9}{16}$	...	...
$\frac{3}{8}$	$\frac{15}{16}$	1	$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{11}{16}$	$1\frac{7}{8}$	$2\frac{1}{8}$
$\frac{1}{2}$	$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{5}{16}$	$1\frac{7}{16}$	$1\frac{11}{16}$	$1\frac{13}{16}$	2	$2\frac{1}{4}$
$\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{5}{16}$	$1\frac{3}{8}$	$1\frac{7}{16}$	$1\frac{9}{16}$	$1\frac{13}{16}$	2	$2\frac{3}{16}$	$2\frac{3}{8}$
$\frac{3}{4}$	$1\frac{7}{16}$	$1\frac{1}{2}$	$1\frac{9}{16}$	$1\frac{5}{8}$	$1\frac{11}{16}$	$1\frac{15}{16}$	$2\frac{3}{16}$	$2\frac{5}{16}$	$2\frac{1}{2}$
$\frac{7}{8}$	$1\frac{5}{8}$	$1\frac{11}{16}$	$1\frac{3}{4}$	$1\frac{13}{16}$	$1\frac{7}{8}$	$2\frac{1}{16}$	$2\frac{5}{16}$	$2\frac{1}{2}$	$2\frac{5}{8}$
1	$1\frac{13}{16}$	$1\frac{13}{16}$	$1\frac{7}{8}$	$1\frac{15}{16}$	2	$2\frac{1}{4}$	$2\frac{7}{16}$	$2\frac{5}{8}$	$2\frac{3}{4}$
$1\frac{1}{8}$	...	2	$2\frac{1}{16}$	$2\frac{1}{8}$	$2\frac{3}{16}$	$2\frac{3}{8}$	$2\frac{5}{8}$	$2\frac{3}{4}$	$2\frac{15}{16}$
$1\frac{1}{4}$	...	$2\frac{3}{16}$	$2\frac{3}{16}$	$2\frac{1}{4}$	$2\frac{5}{16}$	$2\frac{9}{16}$	$2\frac{3}{4}$	$2\frac{15}{16}$	$3\frac{1}{8}$
$1\frac{3}{8}$	...	...	$2\frac{3}{8}$	$2\frac{7}{16}$	$2\frac{1}{2}$	$2\frac{11}{16}$	$2\frac{7}{8}$	$3\frac{1}{16}$	$3\frac{5}{16}$
$1\frac{1}{2}$	...	...	$2\frac{9}{16}$	$2\frac{9}{16}$	$2\frac{5}{8}$	$2\frac{7}{8}$	$3\frac{1}{16}$	$3\frac{3}{16}$	$3\frac{7}{16}$
$1\frac{5}{8}$	...	...	...	$2\frac{3}{4}$	$2\frac{3}{4}$	3	$3\frac{3}{16}$	$3\frac{3}{8}$	$3\frac{9}{16}$
$1\frac{3}{4}$	...	...	...	$2\frac{15}{16}$	$2\frac{15}{16}$	$3\frac{1}{8}$	$3\frac{3}{8}$	$3\frac{1}{2}$	$3\frac{3}{4}$
$1\frac{7}{8}$	...	...	...	...	$3\frac{1}{8}$	$3\frac{5}{16}$	$3\frac{1}{2}$	$3\frac{11}{16}$	$3\frac{7}{8}$
2	...	...	...	...	$3\frac{5}{16}$	$3\frac{7}{16}$	$3\frac{11}{16}$	$3\frac{13}{16}$	$4\frac{1}{16}$
$2\frac{1}{8}$	...	...	...	...	...	$3\frac{5}{8}$	$3\frac{13}{16}$	$3\frac{15}{16}$	$4\frac{3}{16}$
$2\frac{1}{4}$	...	...	...	...	...	$3\frac{3}{4}$	4	$4\frac{1}{8}$	$4\frac{15}{16}$
$2\frac{3}{8}$	...	...	...	...	...	$3\frac{15}{16}$	$4\frac{1}{8}$	$4\frac{1}{4}$	$4\frac{1}{2}$
$2\frac{1}{2}$	...	...	...	...	...	$4\frac{1}{16}$	$4\frac{1}{4}$	$4\frac{7}{16}$	$4\frac{5}{8}$

Table 38. - Length of Rivets for Various Grips (Continued)

Grip	Diameter of Rivet				
	1/2	5/8	3/4	7/8	1
	Hole diameter 1/16 inch greater than rivet diameter				
1/4	1 1/8	1 3/8	1 5/8	...	...
3/8	1 3/8	1 5/8	1 3/4	2	2 1/8
1/2	1 1/2	1 3/4	1 7/8	2 1/8	2 1/4
5/8	1 5/8	1 7/8	2 1/8	2 1/4	2 3/8
3/4	1 3/4	2	2 1/4	2 3/8	2 5/8
7/8	2	2 1/8	2 3/8	2 5/8	2 3/4
1	2 1/8	2 3/8	2 1/2	2 3/4	2 7/8
1 1/8	2 3/8	2 1/2	2 3/4	2 7/8	3 1/8
1 1/4	2 1/2	2 3/4	2 7/8	3	3 1/4
1 3/8	2 3/4	2 7/8	3	3 1/8	3 3/8
1 1/2	2 7/8	3	3 1/4	3 1/4	3 1/2
1 5/8	3	3 1/8	3 3/8	3 1/2	3 5/8
1 3/4	3 1/4	3 3/8	3 1/2	3 5/8	3 7/8
1 7/8	3 3/8	3 1/2	3 5/8	3 7/8	4
2	3 5/8	3 5/8	3 7/8	4	4 1/4
2 1/8	...	3 7/8	4	4 1/8	4 3/8
2 1/4	...	4	4 1/4	4 1/4	4 1/2
2 3/8	...	4 1/4	4 3/8	4 3/8	4 5/8
2 1/2	...	4 3/8	4 1/2	4 5/8	4 3/4

Table 39. Safe Shearing Design Stresses for Rivets

(Aluminum Company of America)

Rivet	Driving Procedure	Safe Shearing Design Stress, Lbs. Per Square Inch
2 S	Cold As-received	3,000
3 S	Cold As-received	4,000
4 S	Cold As-received	5,500
17 S	Cold immediately after quenching	10,000
51 S-W	Cold As-received	8,000
17 S	Hot 930° to 950°F	9,000
51 S	Hot 960° to 980°F	6,000
Steel	Hot 1700° to 1900°F	13,000

20% less for caulked joints.

These values have a factor of safety of about 3.5 based on tests of driven rivets.

Safe Bearing Value of Rivets. - Permanent distortion of rivet holes in aluminum plates begins to take place to any appreciable amount when the bearing stress is about equal to the nominal tensile strength of the material. The ultimate bearing strength amounts to about 1.8 times the nominal tensile strength if the distance from the center of the hole to the edge of the plate is equal to at least twice the diameter of the hole, and decreases in the same proportion as the edge distance is reduced. Values of safe bearing stresses are given in Table 41.

The bearing stresses for rivets (Table 41) are used only when the material of the rivet is softer than that of the plate. For caulked joints the bearing stresses should be reduced about 20 per cent.

In general the following ultimate stresses are applicable to 2 SO aluminum:

Plate:	ultimate tensile strength	13,000	lbs.	per	sq.	in.
	ultimate crushing strength	17,500	"	"	"	"
Rivet:	ultimate strength in single shear	10,000	"	"	"	"

Table 40. Safe Bearing Design Stresses for Various Aluminum Alloys

(Aluminum Company of America)

Alloy	Safe Bearing Design Stress, Lbs. per Square Inch	Alloy	Safe Bearing Design Stress, Lbs. per Square Inch
2 S-0	5,000	A 17 S-T	18,000
2 S-$\frac{1}{2}$ H	7,000	17 S-T	26,000
2 S-H	10,000	24 S-T	29,000
3 S-0	7,000	27 S-T	26,000
3 S-$\frac{1}{2}$ H	9,000	51 S-W	15,000
3 S-H	13,000	51 S-T	21,000
4 S-0	11,000	53 S-W	13,000
4 S-$\frac{1}{4}$ H	14,000	53 S-T	15,000
4 S-$\frac{1}{2}$ H	15,000		
4 S-H	18,000		

20% less for caulked joints.

Table 41. Safe Bearing Design Stresses for Various Driven Rivets

(Aluminum Company of America)

Rivet	Driving Procedure	Safe Bearing Design Stress, Lbs. Per Square Inch
2 S	Cold As-received	7,000
3 S	Cold As-received	9,000
4 S	Cold As-received	14,000
17 S	Cold immediately after quenching	26,000
51 S-W	Cold As-received	15,000
17 S	Hot 930° to 950°F	26,000
51 S	Hot 960° to 980°F	15,000
Steel	Hot 1700° to 1900°F	30,000

20% less for caulked joints.

Design Proportions of Riveted Joints. - Size of Rivet. - The rivet diameter should not be smaller than the thickness of the thickest plate and should not exceed two and one-half to three times the thickness of the plate through which it is driven. A good average value is 2 times the thickness of the plate.

Spacing of Rivets. - Rivets should be spaced not closer than three times their nominal diameter, so that they can be driven without difficulty. The maximum spacing of rivets depends on the type of the joint. It should not exceed twenty-four times the thickness of the sheet. For caulked joints the maximum spacing is four times the rivet diameter or about ten times the thickness of the plate.

Edge Distance. - The edge distance, i.e., the distance from the center of the hole to the edge of the plate should be at least one and one-half times the diameter of the rivet or about four times the thickness of the plate. A good average value is twice the diameter of the rivet.

Welded Joints. - The process most generally used in welding aluminum is fusion welding with an oxy-hydrogen or oxy-acetylene torch. It can be applied to metal of all thicknesses and to cast as well as wrought aluminum.

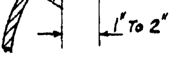


Fig. 112

All customary types of joints, such as butt, lap, tee, fillet joint, etc., can be made. With view to after-treatment of the seam the butt weld should be preferred wherever possible. In Fig. 112 a tank head is welded to a shell by means of a butt joint. Fig. 113 shows how sheets of different thickness are prepared for gas welded butt joints.

Finishing of Welds. - Aluminum welds receive a further finishing other than washing in cases where it is desired to improve the appearance or the strength of the weld. In the first case the greater portion of the

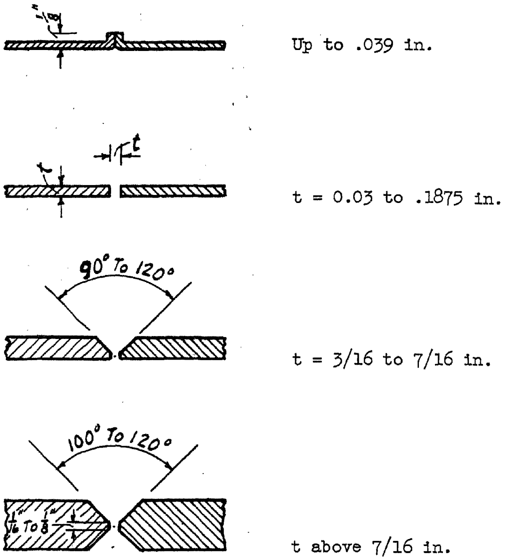


Fig. 113. Preparation of Butt Joints for Torch Welds of Aluminum Sheets.

Note. In plates 1/8" thick and above, notches are to be made with a cold chisel, approx. 1/16" deep and 3/16" apart preparatory to welding.

excess metal is chipped off with a pneumatic chisel. After chipping, the joint is ground to a smooth finish. In the second case the irregular edges of the weld bead are chipped off and then the bead is hammered down to the same thickness as the metal on each side of the weld. Thereby a strong solid seam is produced and also some of the contraction strains set up during the cooling of the weld are relieved.

While the present tendency seems to be to leave aluminum welds wherever possible unfinished it is recommended to hammer welds in 2 S and 3 S alloys on process equipment and pressure vessels which require a maximum strength and frequently also a smooth surface of the weld.

Efficiency of Aluminum Joints. - In unfinished welds, the material is in the as-cast condition and has a slightly lower unit strength than the annealed wrought material adjacent to it. But since the unfinished weld is always built up somewhat, resulting in a larger thickness through the weld than through the plate, it is assumed that the strength of the entire section of welded material is greater than the strength of the section of annealed material adjacent to the weld.

In the case of hammered welds, the thickness through the weld is about the same as through the plate. But, since the material in the weld has been hardened through the hammering, the weld should be at least equal in strength to the annealed material adjacent to the weld.

In both cases, therefore, the strength of the weld can be considered equal to the strength of the wrought material.

The following physical properties of 99.5% pure aluminum in the condition after welding were obtained by tests:

hardness	soft	hard	
elastic limit	2,800- 3,600	5,700- 7,100	lbs. per sq. in.
tensile strength	11,600-12,500	12,800-14,200	" " " "
elongation in 8 in.	20-30	4-8	per cent

Tubular Connections. - Manhole and tubular connections in shells or heads are preferably made by flanging out the wall of the tank and butt welding the nipple to it, as shown in Fig. 114. The joint may be either of the Van Stone type, Fig. 114 or of the fixed flange type, Fig. 115.

If it is not possible to flange the side wall on account of the connection being too small, the tube is inserted in a hole in the side wall and welded from both sides, as shown in Fig. 116. If hardening of the weld is desired the fillet should be made large enough, at least 1" to 2" radius, to permit hammering, Fig. 117. A screwed connection is shown in Fig. 119. The limiting dimensions of tubular connections which can be practically butt welded to a tank are given in Table 42.

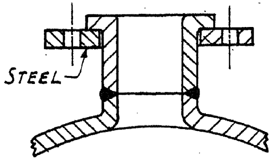


Fig. 114

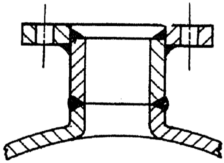


Fig. 115

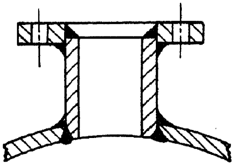


Fig. 116

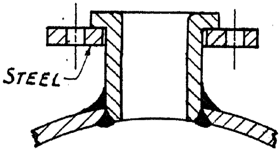


Fig. 117

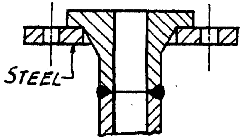


Fig. 118

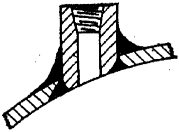


Fig. 119

Table 42. Minimum I.P.S. Tubes that Can Be Butt Welded to Tank Side Walls

(Aluminum Company of America)

Thickness of Side Wall (Inches)	Minimum I.P.S. Tube (Inches)
$\frac{3}{16}$	1
$\frac{1}{4}$	2
$\frac{3}{8}$	4
$\frac{1}{2}$	6

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Chapter XIX

MONEL AND NICKEL VESSELS

Design Formulas. - Wall thicknesses of cylindrical shells and dished heads are calculated by means of the same formulas as used for steel vessels. In cases where the theoretical computed shell thickness is less than 1/8 in., the thickness is often increased, in order to obtain sufficient rigidity. The proper stresses, factors of safety, and joint efficiencies are given below. Values of the t/D ratio for vessels under external pressure may be determined by means of Fig. 4 and 5.

Design Stresses. - The tensile strength of Monel, nickel, and Inconel can be increased by cold working and by addition of strengthening elements. Plates and sheets can, therefore, be furnished to meet specified minimum physical properties. Since the material is usually heated to temperatures above the annealing temperature, calculations should be based on the physical properties in the annealed condition. The International Nickel Company recommends the following stresses:

Table 43. Design Stresses of Nickel Alloys

(International Nickel Co.)

Maximum Temperature Deg. Fahr.	Minimum of Specified Range of the Tensile Strength of the Material in lb./sq. in.			
700°F	65,000	70,000	75,000	80,000
	Maximum Allowable Working Stress			
Monel	......	14,000	15,000	16,000
Nickel	13,000	14,000	15,000	......
Inconel	......	......	15,000	16,000

The A.S.M.E. Code specifies a maximum allowable working stress of 14,000 lbs. per sq. in. up to 406°F for monel.

The modulus of elasticity is:
E = 26,000,000 lbs. per sq. in. for monel.
E = 30,000,000 lbs. per sq. in. for nickel.

The yield strength is:
Sy = 37,500 lbs. per sq. in. for monel.
Sy = 30,000 lbs. per sq. in. for nickel.

Factor of Safety. - The stresses in Table 43 are based on a factor of safety of five which is generally accepted. It is, however, up to the judgment of the designer, to apply a smaller factor of safety where the nature of the design allows a heavier loading. Practical fabricating and design considerations usually call for a larger factor for the thinner gauge materials. The International Nickel Co. suggests the following factors of safety:

3.5 for shells 1/2 in. thick and over.
4 for shells under 1/2 in. thick to 3/16 in.
5 for shells under 3/16 in. thick.

Design of Vessels. - The construction of vessels made of nickel and its alloys is in general similar to the construction of vessels made of other non-ferous metals, such as copper and aluminum except that the greater toughness of the metal requires slightly different dimensions for certain parts. The heads are fabricated in the form of hemispherical, ellipsoidal, and ordinary flanged and dished heads to the same standard dimensions as boiler heads. The free edge of open top vessels is stiffened by simple flanging according Fig. 120. Vessels, having a shell thickness of 3/16 in. or less are usually flanged over a flat iron ring, or an angle iron ring.

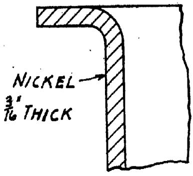


Fig. 120

Bolted covers for atmospheric pressure may be designed without steel backing rings, as shown in Fig. 121.

For higher pressures, steel flanges are required as described for copper vessels. The steel backing rings should be heavier than the sizes used for copper vessels so that they offer greater resistance against deformation when the shell is flanged over and when the bolts are tightened.

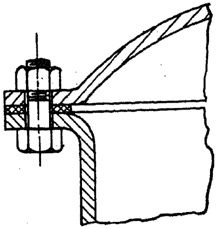


Fig. 121

Riveted Joints. - Riveting is a satisfactory method of joining nickel and monel plates. Rivets 1/2 in. in diameter and under are made cold from bright stock, while sizes above 1/2 in. in diameter are headed hot and, therefore, have a black oxide coat.

Rivets, 5/16 in. in diameter and smaller, are driven cold. Since monel rivets are progressively hardened through the cold driving operation, even to such an extent that the metal becomes too hard to respond to further blows, they should be used in the softest possible temper. Special low-carbon stock is most suitable for this purpose. The plate, on the other hand, should have a high yield point in order to prevent excessive swelling of the rivet hole. The clearance of the rivet hole and the length of the grip should be kept to a minimum.

The proper lengths for various grips of Monel, Inconel, and nickel rivets are the same as those of steel rivets.

The rivet holes should be sub-punched and reamed to size in order to remove material strained in punching, or they should be drilled to size.

Monel, Inconel, and nickel rivets are available in any required size and shape of head. Their weight may be obtained from weight tables for steel rivets by multiplying the values with 1.13.

Caulking. - The caulking is carried out in the same manner and with the same type of tools as for steel plates. The degree of the bevel is 70°. Thin plates are tightened by caulk welding, which should be done after assembling the joint for riveting, but before the rivets are driven. Otherwise, the heat of welding may loosen the rivets.

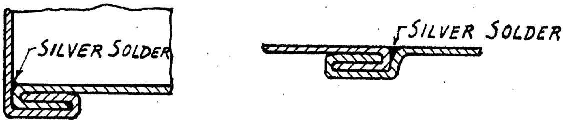
Design of Riveted Joints. - The same design equations, rivet spacing and proportioning of the joint as for steel apply to nickel and its alloys. Minimum safe values of design stresses to be used are given in Table 44.

Table 44. Ultimate Strength Values for Riveted Joints

(International Nickel Company)

	lbs. per square inch			
	Steel	Nickel	Monel	Inconel
Tensile strength of plate	55,000	70,000	75,000	80,000
Shearing strength of rivet in single shear	44,000	47,000	50,000	55,000
Shearing strength of rivet in double shear	88,000	94,000	100,000	110,000
Crushing strength of plate	95,000	100,000	105,000	115,000

Brazed Joints. - Nickel, Monel Metal, and Inconel sheets of No. 18 U.S. Ga. (.0490 in.) and lighter, as used for small storage tanks, are lock seamed and silver soldered. A corner lock and a flush locked seam are shown in Fig. 122. Silver solders flowing at about 1175 degrees F and 1325 degrees F have been successfully employed. The brazing is carried out with a very small oxy-acetylene torch or a lead burner's torch.



a) Folded End Lock

b) Flush Locked Seam

Fig. 122

Welded Joints. - Sheets larger than 0.05 in. thickness and plates are joined by welding. Up to 3/16 in. plate thickness acetylene and atomic hydrogen welding give very satisfactory results, while for heavier plates metallic arc welding is usually employed.

Thin gauge metal is welded by flanging the edges to an angle slightly less than 90°, as shown in Fig. 123, and melting down the turned up edges with the acetylene flame, without employing a welding rod. Crevices in the seam may be filled in with soft or silver solder. One of the applications of the welding method is to join the two half sections of spherical floats.

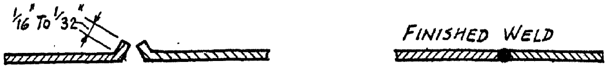
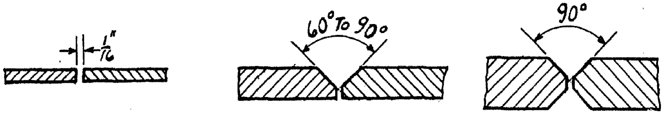


Fig. 123



Up To 5/32"   3/16" To 7/16"   7/16" AND UP

Fig. 124

Butt joints of heavier gauge sheets and plates are prepared as shown in Fig. 124.

In laying-out plates for welding, the plates are spread at the end opposite the start for the weld in accordance with Fig. 125, in order to provide for the expansion and contraction of the metal during welding. The magnitude of the spread which depends on the amount of heat used for welding is approximately 1/8 in. + L x 0.015 for electric welding and 1/8 in. + L x .02 for gas welding if L is the length of the weld.

Tubular Connections. - Screwed Connections are made by welding a heavy spud, Fig. 126, or a welding flange, Fig. 127, of monel or nickel on to the shell.

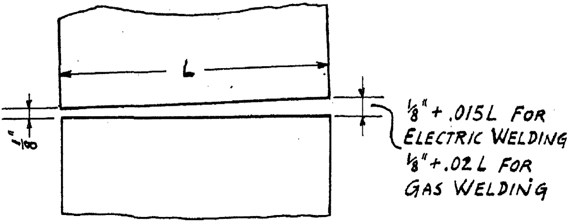

L
1/8"
1/8" + .015L FOR ELECTRIC WELDING
1/8" + .02 L FOR GAS WELDING

Fig. 125

Welding flanges are available in sizes up to 4" I.P.S. Pipes and tubes may be welded on according to Fig. 128 or Fig. 129. Connections in light gauge material are silver soldered as shown in Fig. 130 and Fig. 131.

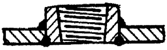


Fig. 126

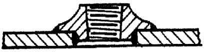


Fig. 127

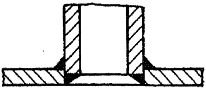


Fig. 128

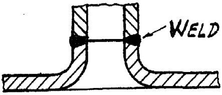


Fig. 129

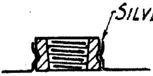


Fig. 130

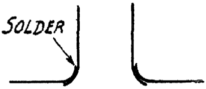


Fig. 131

Pipe Joints. - Light gauge tubing is joined by means of a brazed cup joint, Fig. 132, like copper tubes. Heavy tubes and I.P.S. tubes are welded, Fig. 133.

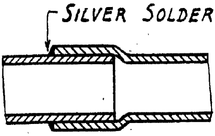


Fig. 132

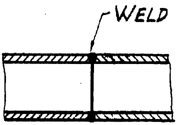


Fig. 133

Flanged joints may be made in several ways. The cheapest type is the loose flange type. In cases like the one illustrated in Fig. 134, a spun tube end is used or the tube is hand flanged over the steel backing ring. In Fig. 135 a loose steel flange is slipped over a Van Stone end which in turn is welded on to a tube or shell as the case may be. Solid monel and nickel flanges are either screwed on, Fig. 136, or welded on, Fig. 137.

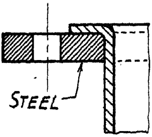


Fig. 134

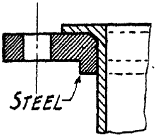


Fig. 135

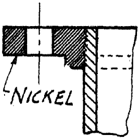


Fig. 136

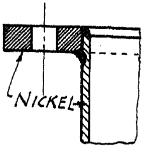


Fig. 137

Pipe Bending. - Cold Bending. - Seamless tubes having a wall thickness of not less than one-fifteenth the diameter, including all iron pipe sizes up to and including 2-1/2 in., can be bent cold to a mean radius of three diameters or greater, without filling. Tubes of thinner gauges should be filled with rosin, lead, or low melting alloys, in preparation for the bending operation. "Annealed" tubing has to be used for cold bending.

Hot Bending. - Seamless tubes above 2-1/2 in. diameter are sand filled and bent hot. The standard minimum radius of the bend is equal to 5 diameters of the tube. However, 2 and 2-1/2 in. I.P.S. tube can be bent hot to a radius as small as two diameters. Special, large diameter, hollow bored tubing up to 6 in., or heavy drawn tubing, having a wall thickness of at least one-twelfth the outside diameter, can be bent hot to four diameters. Tubes used for hot bending are in the "as drawn" temper.

Steel-Jacketed Vessels. - Steel jackets for Monel, nickel, and Inconel vessels can be joined to the inner vessel in several ways. In Fig. 138, the inner shell is flanged over the jacket and closed with a cover.

If the jacket stops below the upper curb ring, the jacket is riveted on to the inner shell in accordance with either Fig. 139 or Fig. 140. When the thickness of the inner shell plate is not less than 3/16 in. and when the stress in the weld does not exceed 6,000 lb./sq. in. the jacket may be welded on as shown in Fig. 141. The space between shell and jacket is usually 1-1/2 to 2-1/2 in. Tubular connections to the inner shell through the jacket may be made by one of the methods illustrated in Figs. 142, 143 and 144.

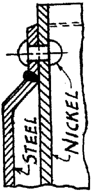


Fig. 138

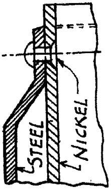


Fig. 139

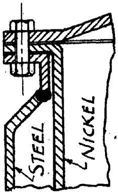


Fig. 140

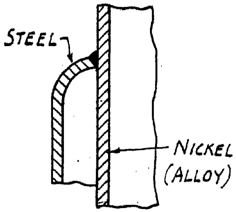


Fig. 141

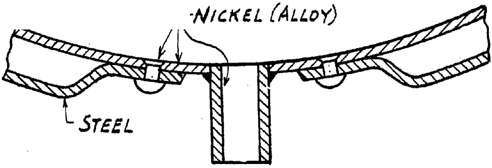


Fig. 142

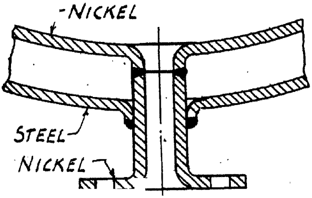


Fig. 143

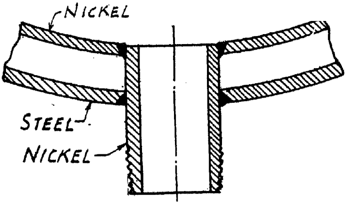


Fig. 144

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Chapter XX

STAINLESS STEEL VESSELS

(18 Chrome--8 Nickel)

Design Formulas. - For the computation of the wall thickness of cylindrical shells and dished heads the same formulas apply as for iron.

Design Stresses. - The design stresses are based on the strength of stainless steel in the annealed condition. The average ultimate tensile strength of annealed stainless steel is 75,000 lb./sq. in. and the yield point 40,000 lb./sq. in. Using a factor of safety of 5 the allowable working stress is 15,000 lb./sq. in. (A.S.M.E. Code). For a factor of safety of 4, the allowable stress becomes 18,750 lb./sq. in. (A.P.I.-A.S.M.E. Code).

Modulus of elasticity E = 29,000,000 lbs. per sq. in.

Design of Vessels. - The construction of stainless steel vessels is similar to the one of nickel vessels. Due to the high cost of the metal heavy stainless steel flanges are avoided and Van Stone type joints with loose carbon steel flanges and steel backing rings are used instead. The greater hardness and strength at high temperatures requires heavier steel backing rings than recommended for copper vessels. The upper curb ring of open vessels is stiffened with iron bars.

Lock-Seamed Joint. - Stainless steel can be soldered if certain precautions are taken to get proper penetration of the solder. Because of its corrosion-resistant nature stainless steel is not affected by soldering fluxes as quickly as ordinary metals. Particularly with polished surfaces the soldering fluid should be given more time to etch the metal. Since these steels have a much lower thermal conductivity than wrought iron and other metals and since a little longer time is required to bring the metal to be soldered up to the temperature where the solder will flow, it is preferable to use large soldering irons heated to a moderate temperature than smaller irons heated to higher temperatures. Buckling during soldering which is caused by the high coefficient of expansion of stainless steel can be avoided through the use of copper chill plates clamped to the work.

Due to the fact that the higher tensile strength of stainless steel tends to cause a springback at a joint, while the high coefficient of expansion of stainless steel tends to break the solder from the metal during expansion and contraction in service, it is not desirable to depend on soldering for strength of the joint. In contrast to copper which will lie perfectly flat when turned over in a lock seam, stainless steel, due to its much higher strength and work hardening characteristics, tends to spring slightly so that the solder breaks away from the metal after it has been in service for some time. For this reason it has been found that lock-seamed and soldered joints do not remain tight over a long period. In order to overcome this difficulty, spot welding or riveting has been applied successfully to strengthen the joint, with soldering used to produce the sealing effect.

Silver Soldering - Stainless steel can be silver soldered to copper and to stainless steel. The latter operation, however, requires considerable skill on the part of the operator.

Brazing. - Brazing of stainless steel is not recommended because the ease of welding this metal and the possibility of inter-crystalline penetration of the brazing alloy which lowers corrosion-resistance under wet corrosive conditions by favoring electrolytic action, makes brazing unadvisable. Embrittlement is also produced if the brazing alloy penetrates into the grain boundaries of the metal.

Welded Joints. - Stainless steel may be welded by either the oxy-acetylene gas torch or electric arc.

It may also be seam, spot and resistance welded, but cannot be forge or hammer welded. In welding stainless steels it is important to keep in mind that their coefficient of expansion is 50% greater than that of mild steel, and that the thermal conductivity if about 50% less. Their electrical resistance in comparison to mild steel is about 6 times as large in the annealed condition and about 12 times as large after cold working.

Gas welding is more suitable for the welding of light gauges, No. 16 (.062) gauges and lighter. Electric welding, however, is generally more satisfactory than gas welding because it imparts less heat to the metal and there is no danger of carbon pick-up as in the case of the gas flame. Low carbon rod coated with non-carbonaceous material should be used. Recently alternating current arc welding machines have been perfected to allow welding of gauges as light as No. 26 gauge (0.018).

Welds made with rods of the same material remain ductile and tough and have a high resistance to shock because this alloy does not respond to hardening by heat treatment.

The corrosion resistance of welded stainless steel is generally satisfactory. However, if the metal is in contact with an active corrosive agent local attack takes place due to the difference in structure between the weld and the adjacent metal which is caused by carbide precipitation during the welding operation. Welded stainless steel equipment, therefore, should be annealed after welding, or fabricated from a material which will retard the precipitation of carbides during the welding operation, if it is to be exposed to severe corrosive attack.

Riveted Joints. - Stainless steel may be riveted hot or cold. Cold riveting is in general limited to sizes less than 3/16 in. and should be executed with a single heavy blow of the hammer, because of the work hardening properties of the metal. Larger rivets are heated to a temperature of 2100°F. They should be set quickly so that the mechanical deformation is completed before the temperature of the rivet drops below 1750°F. The equipment should then be annealed if it is subjected to severe corrosive conditions.

The following ultimate strength values are suggested for the design of riveted joints of 18-8 Stainless Steel:

Plate: tensile strength 80,000 lbs. per sq. in.
crushing strength 110,000 lbs. per sq. in.
Rivet: single shear 55,000 lbs. per sq. in.
double shear 110,000 lbs. per sq. in.

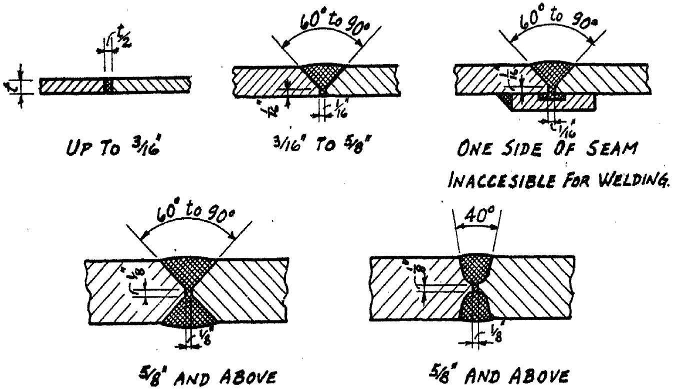
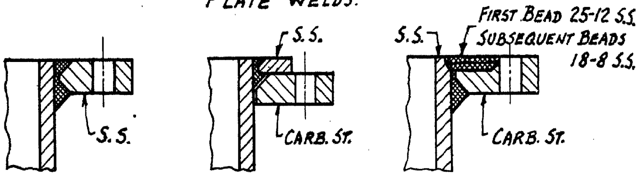
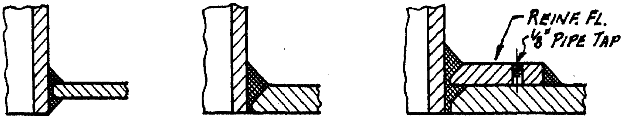


PLATE WELDS.



FLANGE WELDS.



NOZZLE WELDS.

Fig. 144a. Welded Stainless Steel Joints.

Chapter XXI

VESSELS OF ALLOY-CLAD STEEL

Introduction. - Alloy-clad steel consists of a light layer of corrosion resistant material permanently bonded to a heavier layer of steel, by means of hot-rolling. The most important types are Nickel-Clad, Copper-Clad, and Stainless-Clad Steel. Nickel has the advantage that its coefficient of heat expansion is approximately the same as that of steel. The thickness of the nickel normally amounts to 10% and that of the stainless veneer to 20% of the total plate thickness. Plates with a lighter or heavier cladding can be furnished. Through this combination savings of from 1/3 to 1/2 in cost over solid corrosion-resistant materials are effected.

Clad steel can be bent and formed without separating the two metals. It is available in practically all spun and pressed shapes in which ordinary steel plates can be obtained, such as flanged heads, flanged and dished heads, elliptical heads, flued openings for manholes, etc.

Strength of Clad Steel. - In stress calculations the thickness of the mild steel only is taken into consideration. The steel layer is normally low carbon flange quality steel, with a minimum tensile strength of 55,000 lbs. per sq. in. The yield point is 27,500 lbs. per sq. in. and the modulus of elasticity of the composite metal is 30,000,000 lbs. per sq. in.

Design formulas and data used are the same as for steel.

Design of Vessels. - The design follows closely the general practice in designing similar equipment in steel except that here particular attention has to be given that a continuous surface of the corrosion-resistant material on the exposed side is obtained.

The top edge of open vessels in lighter plate gauges is flanged as shown in Fig. 145. A closed pressure or vacuum vessel is shown in Fig. 146.

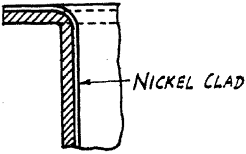


Fig. 145

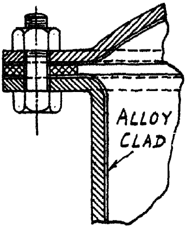


Fig. 146

Welded Joints. - Nickel- and Stainless-Clad steel can be welded without great difficulty by an experienced operator by means of the oxy-acetylene torch as well as the electric arc. The metallic arc method is generally applied to heavy steel plate, while the carbon arc is best suited for welding light gauge linings in nozzles and manholes and other sheet metal in connection with plate work.

The weld on the clad side is preferably completed first, using a rod of the same material as the clad material, i.e., nickel or stainless steel. After cleaning the root of the bevel the weld on the steel side is made.

Various types of joints in Nickel-Clad Steel are shown in Figs. 147 to 149. The same design applies, of course, to stainless-clad steel, if stainless material is substituted for nickel. The butt weld, Fig. 147, should be preferred, wherever possible. Methods of beveling the edge of the plate are indicated in Fig. 150. A variation of the butt joint is the strap joint, illustrated in Fig. 148. The lap joint in Fig. 149 is reinforced with a plug weld.

An inside fillet weld is shown in Fig. 151, and a double fillet or Tee-weld in Fig. 152. In the double fillet weld the clad material is removed so that the weld can penetrate into the steel.

Fig. 153 shows various types of outside corner

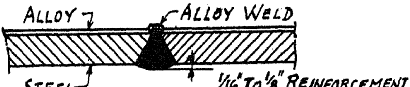


Fig. 147. Butt Joint.

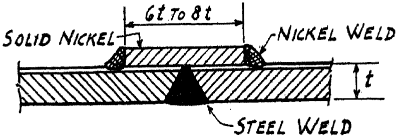


Fig. 148. Strap Joint.

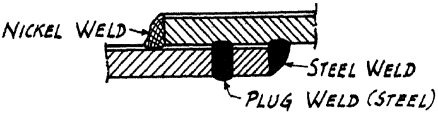


Fig. 149. Lap Joint.

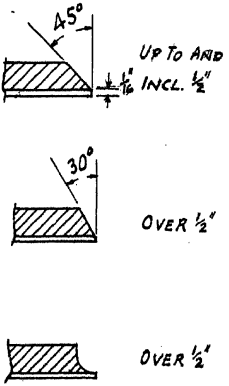


Fig. 150. Preparation of Nickel-Clad Steel Plates for Welding.

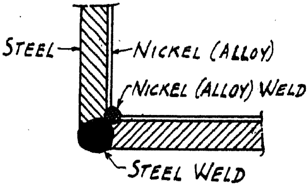


Fig. 151. Inside Fillet.

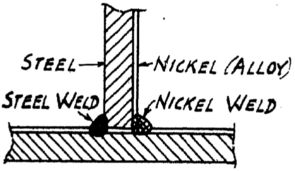


Fig. 152. Double Fillet.

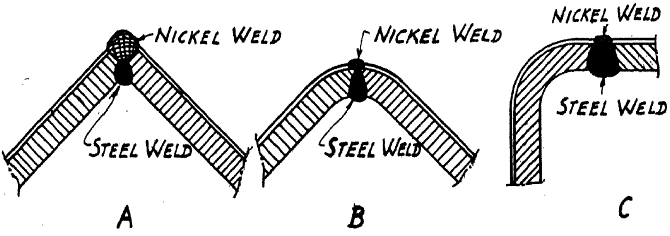


Fig. 153. Outside Corner Weld.

welds. Type A is not recommended because the steel penetrates into the nickel weld and lowers the corrosion resistance.

Riveted Joints. - Clad steel plates can be joined by means of riveting, but riveted joints in general do not offer any advantages over welded joints and are higher in cost. They are necessary in certain cases, f.i., in the construction of railroad tank cars, to meet the requirements of codes and regulations.

The single or double riveted lap joint, Fig. 154, is the type most frequently used. The material of the rivets is the same as the clad material, and the same precautions have to be taken in driving the rivets as recommended for nickel, resp. stainless steel rivets.

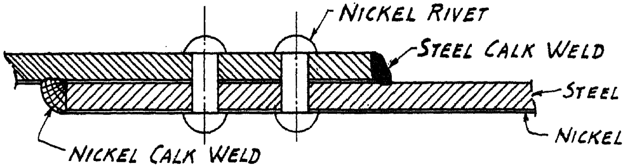


Fig. 154. Double Riveted Lap Joint.

The Butt Strap Joint, with nickel straps against the clad surface is rather high in cost. It is, therefore, used only when special conditions require it.

The design and proportioning of riveted joints, clearances of rivet holes, etc., are the same as for steel plate.

Tubular Connections. - Tapped connections are shown in Fig. 155 to Fig. 157. Small connections, not

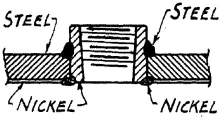


Fig. 155

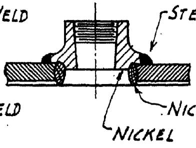


Fig. 156

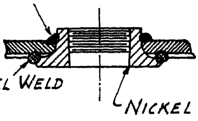


Fig. 157

exceeding 2" I.P.S. may be cut from nickel or stainless bar stock, Fig. 155. The tapped welding flanges, shown in Fig. 156 and 157, are furnished in solid nickel up to 4 in. I.P.S.

For flanged connections up to approximately 6" I.P.S., solid nickel resp. stainless steel tubes are welded to the shell. The free end of the nozzle may either be equipped with a welding flange of solid nickel, Fig. 158, or nickel clad steel, Fig. 159, or it may be flared over a steel backing flange, Fig. 160. In Fig. 161, a Van Stone end in nickel with steel flange is used.

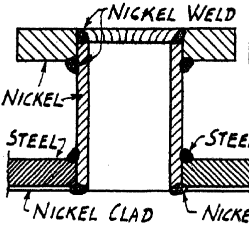


Fig. 158

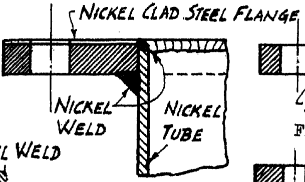


Fig. 159

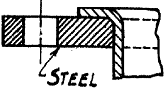


Fig. 160

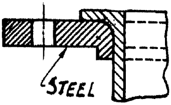


Fig. 161

Large connections, handholes and manheads are built-up of clad steel, Fig. 162, or of steel plate lined with light gauge nickel resp. stainless sheet, Fig. 163 and Fig. 164. Commercial sizes of manholes and manhole covers of nickel clad steel are shown in Fig. 165.

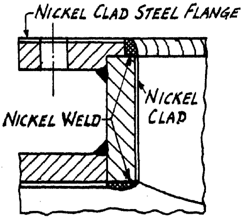


Fig. 162

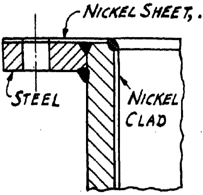


Fig. 163

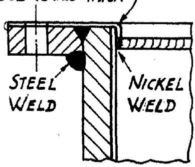
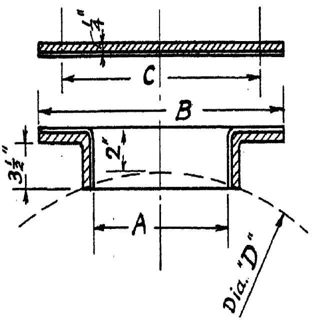


Fig. 164



Manhole Size	A	B	C	D
16"	16"	20½"	19	47
18"	18"	22½"	21	58
20"	20"	24½"	23	81

Fig. 165. Dimensions of Nickel-Clad Steel Manholes. (International Nickel Company).

Jacketed Vessels. - Steel jackets are welded to the clad steel as per Fig. 166. For small connections a nipple of solid corrosion-resistant material is welded to the inner shell. Larger connections are treated similarly except that one of the tubular connections described previously is substituted for the nipple.

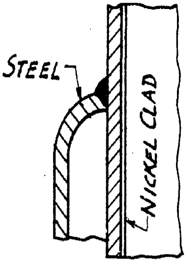


Fig. 166

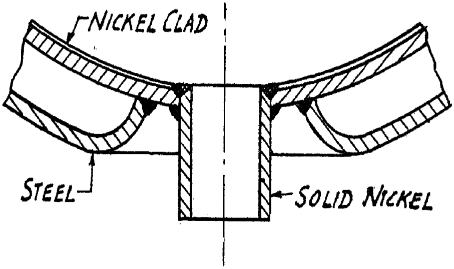


Fig. 167

Chapter XXII

SHELL STRESSES DUE TO DEAD AND WIND LOAD

Wind Load. - Towers and vertical drums are subjected to a wind load, if they are placed outside of buildings. The unit pressure exerted by the wind varies with local conditions and the form of the exposed surface. For American conditions and cylindrical bodies it is customary to use a unit wind pressure of 25 lb. per sq. ft. The exposed surface is represented by the projected area of the tower. For elliptical heads a rectangular area is usually substituted which has a height approximately equal to 3/4 of the depth of the dish. If the tower is insulated the outside dimensions of the insulation are applied. If many large manholes, ladders and other structures are attached to the tower, the projected area which has the largest value should be used. Usually an approximate allowance of one foot is made on each side. If the drum is supported by lugs fastened to the shell near the bottom, the exposed area below the level of support is neglected.

The total wind load is equal to the product of the projected area and unit wind pressure. It is considered as acting at the center of gravity of the projected area.

The total wind pressure in pounds acting above any section at a distance h below the top of the drum is

$$P = D \cdot h \cdot p \qquad (1)$$

where D = outside diameter of exposed surface, ft.
h = distance of any point below the top of drum, ft.
p = unit wind pressure, lb. per sq. ft.

Stresses Due to Wind Load. - The bending moment in inch-pounds at any distance h below the top of the drum is

$$M = \tfrac{1}{2} \cdot p \cdot D \cdot h^2 \cdot 12 \qquad (2)$$

The unit stress in the extreme fiber, in pound per square inch, due to the wind is

$$S_w = M \cdot c/I \tag{3}$$

where c equals the outside radius of the shell in inches, and I equals the moment of inertia of the shell in inches^4; for cylindrical shells $I = \pi(r_1^4 - r_2^4)/4$.

For shells with a comparatively small r/t ratio we can set approximately $I = \pi r^3 \cdot t$. Since r is given in ft. and t in inches, $I = 1728\pi r^3 \cdot t$.

Substituting M, c, and I the unit stress in the extreme fiber of the shell becomes

$$s_w = \frac{0.0533p \cdot h^2 \cdot D}{t \cdot d^2}, \text{ lb. per sq.in.} \tag{4}$$

where d = outside diameter of shell, ft.

For plain uninsulated drums, where D equals d, the unit stress in the shell is

$$s_w = \frac{0.0533\, p \cdot h^2}{t \cdot d} \tag{4a}$$

For p = 25 lb. per sq. ft.

$$s_w = \frac{1.33\, h^2}{t \cdot d} \tag{4b}$$

(t in inches, h and d in ft.)

The stress per lineal inch along the circumference is

$$S_w = \frac{0.0533\, p \cdot h^2 \cdot D}{d^2}, \text{ lb. per lin. in.} \tag{5}$$

respectively

$$S_w = \frac{0.0533\, p \cdot h^2}{d} \tag{5a}$$

For p = 25 lb. per sq. ft.

$$S_w = \frac{1.33\, h^2}{d} \tag{5a}$$

The stress per lineal inch on any circumference can also be computed by dividing the moment M by the area of the stack. Thereby, the same formula is obtained as above

$$S = \frac{0.0533\ p \cdot h^2}{d}\text{, lb. per lin. in.} \tag{5a}$$

The equation for the shell thickness in inches required for a cylindrical shell subjected to wind load only is derived from equation (4a)

$$t = \frac{0.0533\ p \cdot h^2}{d \cdot S_t \cdot E} \tag{6}$$

where S_t = allowable unit stress on the net section, lb. per sq. in.

E = efficiency of the joint.

For p = 25 lb. per sq. ft.

$$t = \frac{1.33\ h^2}{d \cdot S_t \cdot E} \tag{6a}$$

Stresses Due to Dead Loads. - Through the weight of the shell certain longitudinal stresses are produced in vertical drums, which are computed by dividing the weight by the area of the cross-section of the shell. Accordingly the unit stress in lb. per sq. in. due to the weight of the drum above any section is

$$S_1 = \frac{W_1}{12\pi d \cdot t} \tag{7}$$

where W_1 = weight of shell above the section, d = diameter of shell in ft., t = thickness of shell in inches.

For shells with open ends, like stacks, which have a constant thickness above the horizontal joint, the unit stress is approximately

$$s = 3.4\ h \tag{7a}$$

where h = height of the shell above the section, ft.

The load per lineal inch on any circumference of the shell due to its weight is

$$S_1 = \frac{W_1}{12\pi \cdot d} \qquad (8)$$

The stress due to the weight of the shell is comparatively small. It is, therefore, usually neglected for towers, and for stacks less than 200 ft. high.

In a similar way, the load on horizontal joints due to the weight of attachments, internal equipment (bubble cap plates in towers, insulating material, fire brick linings in stacks, etc.), is computed by substituting these loads in equations (7) and (8).

The load per lineal inch of circumference of the shell due to weight of the lining and other attachments, therefore, will be

$$S_2 = \frac{W_2}{12\pi \cdot d} \qquad (9)$$

The weight of the lining and attachments is omitted in calculating the maximum tensile stresses because the maximum load may occur during the erection of the shell before the lining has been installed.

Combined Wind and Dead Load. - The maximum compressive stress on any circumference of shells, subjected to wind and dead loads, is

$$\text{max. } S_{\text{compr.}} = S_w + S_1 + S_2 \qquad (10)$$

Under the same loading, the maximum tensile strength on any circumference of the shell is

$$\text{max. } S_{\text{tens.}} = S_w - S_1 \qquad (11)$$

The number of rivets required in a circumferential joint of a stack is

$$n = \frac{(S_w + S_1 + S_2)\pi \cdot d}{V} \qquad (12)$$

where V = shearing value of one rivet or the bearing value of the plate, whichever is the least.

The minimum thickness of a welded circumferential joint (throat dimension for fillet welds) is

$$t = \frac{(S_w + S_1 + S_2)}{E \cdot s} \tag{13}$$

where S = allowable unit shearing stress of the weld, in lb. per sq. in. and E = efficiency of the joint.

Combined Wind Load and Internal Pressure. - In vertical drums and towers under operating conditions, which are subjected to wind load as well as internal pressure, additional tensile stresses are set up by the hydrostatic end pressure. On the windward side of the shell these stresses tend to amplify the stresses produced by the wind load, while on the lee side they are counteracting the compressive stresses and, therefore, have to be deducted.

Then, if S_p denotes the longitudinal stress due to the internal pressure (Eq. 1, Chap. VI), the maximum compressive stress, in lb. per sq. in., in any horizontal section of the shell is

$$\max.\ s_{\text{compr.}} = s_w + s_1 + s_2 - s_p \tag{14}$$

Under the same conditions, the maximum tensile stress, in lb. per sq. in., is

$$\max.\ s_{\text{tens.}} = s_w - s_1 - s_2 + s_p \tag{15}$$

The stresses due to the weight of the shell and other dead loads are relatively small and can in most cases be neglected, so that for practical purposes the maximum stress in the shell is

$$\max.\ s_{\text{tens.}} = s_w + s_p \text{ (approx.)} \tag{15a}$$

If the total stress exceeds the allowable stress, the wall thickness has to be increased beyond that required by the hoop stresses. Since in a drum under internal pressure the longitudinal stresses are only half as large as the hoop stresses (Eqs. 1 and 2, Chap. VI), the total stress will in most cases be smaller than the allowable stress. However, for high towers it is frequently necessary to increase the wall thickness in the lower portion. Assuming that half the thickness of the shell after corrosion is available to carry the wind load, the length of the shell in ft. below the top of the tower, which does not have to be increased in thickness, is from (6) approximately

$$h = \sqrt{\frac{t \cdot d \cdot s \cdot E}{0.1066\ p}} \tag{16-}$$

The thickness of the increased portion of the shell can be computed by figuring separately the thicknesses required to stand wind load and hydrostatic end pressure and adding them.

Stresses in Vertical Columns Due to End Forces. - If heavy loads, such as overhead condensers and agitator drives, are placed on top of comparatively thin and slender vertical drums, local buckling may occur if the compressive stresses exceed the yield point of the material. For a check on these stresses the following formulas are tentatively given. The theoretical formula for the critical stress at which buckling occurs in thin cylindrical tube subjected to a compressive end force is

$$s' = \frac{E}{\sqrt{3}\sqrt{1-\mu^2}}\frac{t}{R} \tag{17}$$

where R is the mean radius of the tube and μ equals Poisson's ratio. On account of imperfections in the material and the shape of the tubes, the actual critical stresses amount to only 40 to 60 per cent of the theoretical value depending on the R/t ratio. J. R. Younger (Ref. 4) accordingly proposes the following conservative formula which equals approximately 20% of the theoretical value

$$s = 0.12\ E\ t/R \tag{18}$$

R. J. Rowark (Ref. 2) gives the following formula for the safe unit load in large fabricated tubular columns of structural steel:

$$\frac{Q}{A} = XY \tag{19}$$

where $Y = 1$, for $L/R = 60$ and under

$$Y = \frac{21,600}{18,000 + (L/r)^2}, \text{ for } L/R \text{ more than } 60$$

$$X = [1,000,000\ t/R][2 - \frac{2}{3}(100\ t/R)] \text{ for } t/R \text{ up to } 0.015$$

$$X = 15,000 \text{ for } t/R \text{ above } 0.015$$

Minimum $t = 1/4$ in.
Q = load, lb.
A = cross-sectional area of tube, sq. in.

REFERENCES

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Chapter XXIII

SUPPORTS

Introduction. - Tanks with flat bottoms are placed on flat steel plates, concrete slabs or sand cushions. Small containers frequently receive a base ring of flat iron or angle iron. When it is desired to control corrosion from the ground or when the bottom is sloped, larger tanks are set upon dunnage beams (I-beams, channels, or compound sections) which are arranged parallel to the smaller side of rectangular tanks or radially in the case of cylindrical tanks. The principal methods of supporting horizontal cylindrical vessels are saddles and lugs. Vertical drums with dished bottoms are supported by means of legs, lugs, and skirts.

Design of Saddle Supports. - Horizontal drums are supported either directly by concrete piers (Fig. 168) whose top is shaped to the outside radius of the drum or by saddles of steel (Fig. 169) or cast iron which rest upon a level foundation.

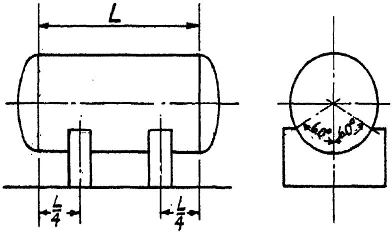


Fig. 168

The factors which govern the design of saddles are the stresses in the shell of the drum, the bearing pressure of the foundation, and the stresses in the iron saddle. If the saddles are placed too far apart and if

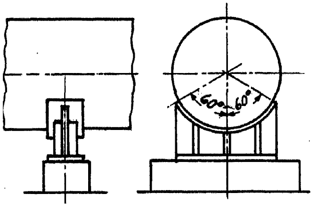


Fig. 169

the contact area between saddle and drum is too small, excessive stresses may be set up in the shell which cause a local buckling or rupture of the shell. The steel saddle is principally checked regarding the compression in a horizontal section, by dividing the load per saddle by the area of the horizontal section. The bearing pressure upon the foundation is obtained by dividing the load per saddle by the area of the base plate. Both the unit compression in the saddle and the bearing pressure upon the foundation have to remain within the allowable limits.

The saddle supports for small tanks are usually placed at the quarter points. For longer tanks which require more than two supports, the center to center distance of the supports usually does not exceed 10 ft. The angle of contact between drum and saddle should not be less than 120°.

Design of Leg and Column Supports. - Legs of drums are usually made of angle iron, tees, channels, H-beams, and similar structural steel sections. They are fastened either directly to the shell of the drum by riveting or welding (Fig. 170), or they are bolted to angles and plates, which are welded to the shell (Fig. 171). Kettles of light wall construction are supported at the steel backing ring of the flanged joint, and secured by a flat iron ring with spanner bolt around the lower part of the shell (Fig. 172).

Compression. - The legs and their attachment to the shell have to be strong enough to carry the weight of the full drum. If the drum is installed outside of a building, the wind load has to be added. The total

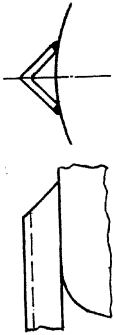


Fig. 170

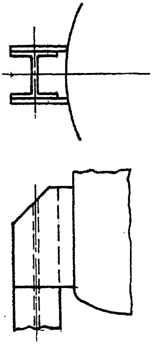


Fig. 171

wind load equals the unit wind pressure times the exposed area and is assumed to be acting at the center of gravity of the exposed area. The moment P · h produced by the wind load tends to overturn the drum around axis A-B, while the weight of the drum tends to hold it down on its foundation (Fig. 173). The highest compressive stresses in the supports occur on the leeward side, when the drum is full, as here the stresses due to dead load and wind load are added. The highest tensile stresses are set up on the windward side when the drum is empty, as here the dead load is subtracted from the wind load.

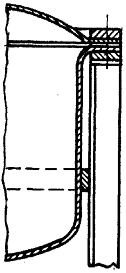
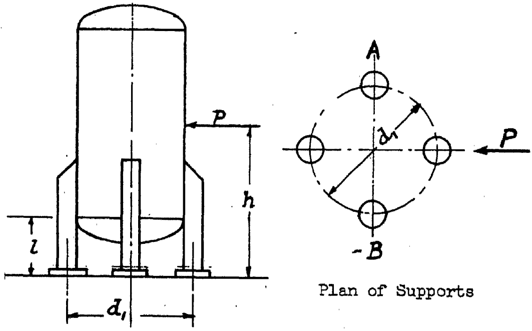


Fig. 172



Plan of Supports

Fig. 173

From this, it will be seen that the stresses on the leeward side are the determining factor for the design of the legs. The maximum total compression, in pounds, in the most remote column is

$$C = \frac{4P \cdot h}{n \cdot d_1} + \frac{W_1 + W_2}{n} \qquad (1)$$

where P = total wind load on exposed surface, lb.
h = vertical distance of center of gravity from exposed surface to top of foundation, ft.
d_1 = diameter of anchor bolt circle, ft.
n = number of legs
W_1 = weight of empty drum, lb.
W_2 = weight of liquid and other dead load, lb.

If the leg is attached to the drum, so that it can be considered as a column under concentric end load, the allowable fiber stress, in lb. per sq. in., is

$$f = \frac{18000}{1 + \frac{1}{18000}\left(\frac{L}{r}\right)^2} \qquad (2)$$

where L = unbraced length of leg, in.
r = least radius of gyration of leg, in.

The maximum permissible L/r ratio is 120, and the maximum fiber stress = 15,000 lb. per sq. in.

The cross-section of the column, in sq. in., is

$$A = C/f \qquad (3)$$

where C = maximum compression per leg, lb.

Eccentric Loading. - If the attachment of the leg to the drum is such that bending stresses are produced through eccentricity, in addition to the axial compression, the allowable fiber stress, in lb. per sq. in., is

$$f = \frac{C}{A} + \frac{C \cdot e \cdot c \cdot}{I} \qquad (4)$$

where e = eccentricity (distance center line of shell to center line of leg), in.
c = distance of extreme fiber to neutral axis of leg, in.
r = least radius of gyration of leg, in.

Direct Bending. - The legs are also subjected to direct bending by the wind load. For calculating purposes, two legs and the drum are considered as forming a rectangular frame with rigid corners, and the upright members as being firmly anchored in the ground. The wind acts at the upper end of the unbraced length of the legs and tends to deflect them. The maximum fiber stress, in lb. per sq. in., is

$$f = \frac{P/n \times L/2}{S} \tag{5}$$

where P = wind load, lb.
L = unbraced height of leg, in.
n = number of legs
S = section modulus of leg (mean value of S_x and S_y)

The joint between shell and leg has also to be checked with regard to the shear stresses.

Anchor Bolts of Column Supports. - The stresses in anchor bolts depend on the wind load and the weight of the drum. They reach their highest value in the supports on the windward side when the drum is empty. Here, the maximum anchor bolt load, in lb., per support is

$$T = \frac{4P \cdot h}{n \cdot d_1} - \frac{W_s}{n} \tag{6}$$

Notations are the same as above. A positive value of T indicates that the drum is held on the foundation by its own dead weight. The bolt area at the root of the thread, in square inches, required per support is

$$A = T/s_b \tag{7}$$

where s_b = allowable unit tensile stress in the bolt, in lb. per sq. in.
= 9,000 for pre-stressed carb. steel bolts,
= 11,000 for not pre-stressed bolts,
= 16,000 in structural steel design.

The length of imbedment of the bolts in the foundations, in inches, is

$$L = \frac{T}{3.14\ u \cdot d} \qquad (8)$$

where d = diameter of bolt, in.
u = allowable unit bond stress, lb. per sq. in.

If sleeves are used with the bolts, they should have a diameter at least one inch larger than that of the bolts, and their length should be about 20 bolt diameters.

Base Plate of Column Supports. - The maximum total compression C per column is from equation (1)

$$C = \frac{4P \cdot h}{n \cdot d_1} + \frac{W_1 + W_2}{n}\ , \text{ lbs.} \qquad (1)$$

Notations being the same as above. The load acts on a small area of the base plate and is transmitted through the plate to the foundation. In the case of an H column, it is assumed that the column load is distributed within a rectangular area whose dimensions are 0.95d and 0.8b (Fig. 174), where d = depth and b = width of the column. It is also assumed that the base plate has a uniform bearing pressure, p, on the foundation.

The area of the base A, in sq. in., is then computed from the unit bearing pressure, p, in lb. per sq. in., allowed on the type of masonry to be used for the foundation, and the load on the column C.

$$A = C/p \qquad (9)$$

The dimensions B and C, in inches, are then determined so that the edge distances e and f (Fig. 174)

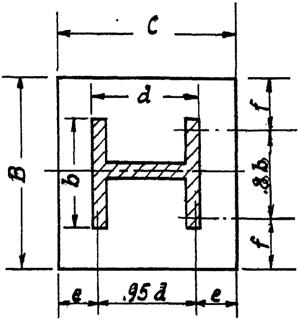


Fig. 174

are approximately equal and so that B x C larger than A. Assuming an allowable stress of 18,000 lb. per sq. in. for the steel slab, the thickness of the plate, in inches, is computed from

$$t^2 = p \cdot e^2/6000 \qquad (10)$$

or if f is greater than e

$$t^2 = p \cdot f^2/6000 \qquad (11)$$

For the unit bearing pressures of steel plates upon masonry the following maximum values are used, in lb. per sq. in.

Brick masonry laid in cement mortar	200
Portland cement concrete, 1-2-4	500
First-class sandstone	300
First-class limestone	400
First-class granite	600

Lug Supports. - Where a vertical drum is supported on a steel structure, lug supports are frequently used. The maximum dead load and wind load acting on one lug is computed in the same manner as for leg supports. Angle iron, tees and H-beams equipped with a proper base plate are used as lugs on small drums. For heavy loads, the lugs are made up of steel plates. They may be riveted or welded to the drum. For riveted construction, cast iron or cast steel lugs are also applied. The vertical plate by which the lug is attached to the drum should be sufficiently large, and a sufficient number of lugs should be provided so that excessive local stresses are avoided in the shell of the drum.

The joint between lug and shell is subjected to vertical shear and to a bending moment due to the eccentric load.

Design of Riveted Lugs. - For the design of riveted lug joints (Fig. 175) the same method is used as for the design of eccentric brackets for beam supports.

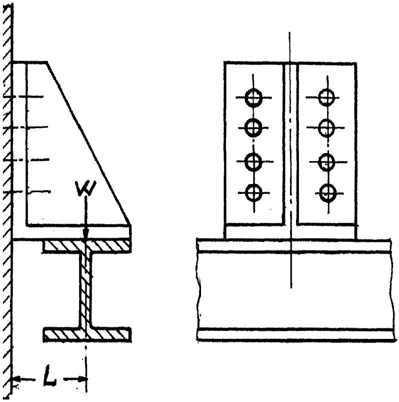


Fig. 175

For a joint with two rows of rivets the following equation exists

$$W = 2\,C \cdot S \qquad (12)$$

where W = total load per lug, lbs.
S = allowable stress on one rivet, lb. per sq. in.
C = coefficient, as given below

The coefficient C is computed as follows

$$C = \frac{n}{\sqrt{\left[\frac{6\,L}{(n+1)b}\right]^2 + 1}} \tag{13}$$

where n = total number of rivets in any one vertical row
L = eccentricity of load, in.
b = pitch of rivets, in.

The American Institute of Steel Construction (Ref. 2) has compiled a table of the coefficients C for various values of n and L. These coefficients are computed for a single row of rivets and should be multiplied by 2, if two rows of rivets are used. They are based on a pitch b = 3 in. A simple procedure is to calculate the coefficient C = F/2 S and then to select the nearest value of n in the proper "L" column of the table.

Design of Angle Iron Brackets. - Where short pieces of angle iron are welded as brackets to a drum or tank (Fig. 176), the length and thickness of the angle has to conform to the following equation.

$$b \cdot t^2 \geqq \frac{W}{10}\left(k + 1 - \frac{2t}{3}\right) \tag{14}$$

where b = length of bracket, in.
t = thickness of angle, in.
W = total load per bracket, lb. per sq. in.
k = minimum length of effective bearing, in.

For the leg dimension a and weld size d the following relation exists

$$d \cdot a \geqq \frac{W}{19.2}\sqrt{1 + [1.5(k+1)/a]^2} \tag{15}$$

where a = length of attached leg of angle, in.
d = size of weld, in.

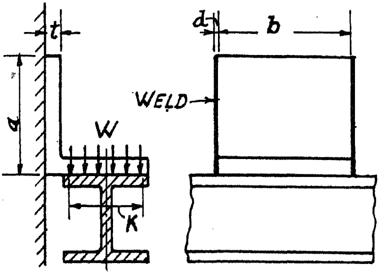


Fig. 176

Design of Welded Lugs. - The welded joint between lug and shell is subjected to shear as well as bending stresses due to the eccentricity of the load. The resultant of both forces is, therefore, the determining factor for the strength of the weld. In the calculation first the center of gravity of the weld is determined and then its polar moment of inertia. For this purpose, the weld is usually treated as a line and the working stress in terms of lb. per lineal inch is used. For a single web bracket (Fig. 177) the vertical force, in lb. per lin. in., is

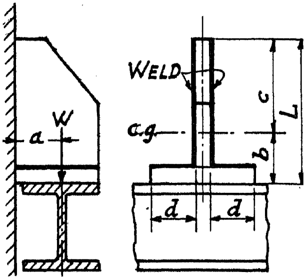


Fig. 177

$$\frac{W}{2(L + d)} \qquad (16)$$

where W = beam reaction, lb.
L = height of bracket, in.
d = length of one horizontal leg, in.

The horizontal force is

$$H = W \cdot a \cdot b / I_o \tag{17}$$

where a eccentricity of load W, in.
b vertical distance of center of gravity to level of support, in., $= L^2/2(L + d)$
I_o = polar moment of inertia of weld, in.^{4}

The polar moment of inertia for 2 welds of this type is

$$I_o = \frac{b^3 + c^3}{1.5} + 2d \cdot b^2$$

where $c = L - b$

The resultant unit working stress in the weld, in lb. per lin. in., is

$$R \quad \sqrt{V^2 + H^2} \tag{18}$$

The stresses in the welds of lugs attached by two vertical fillet welds are conveniently computed by means of Fig. 178. The coefficient K, corresponding to a certain L/a ratio, is plotted from the curve and inserted in the equation for the maximum actual load per linear inch of weld, in lb.

$$R_n = W K/L \tag{19}$$

The load obtained is for the two sides. The allowable stress, in lb. per sq. in., equals the allowable working stress of the material times the joint efficiency (fillet weld) times the side weld factor times the shear factor. For fire-box quality steel (TS = 55,000) the allowable unit stress is

13,750 x .65 x .75 x .8 = 13,750 x .39 = 5,360 lb. per sq. in.

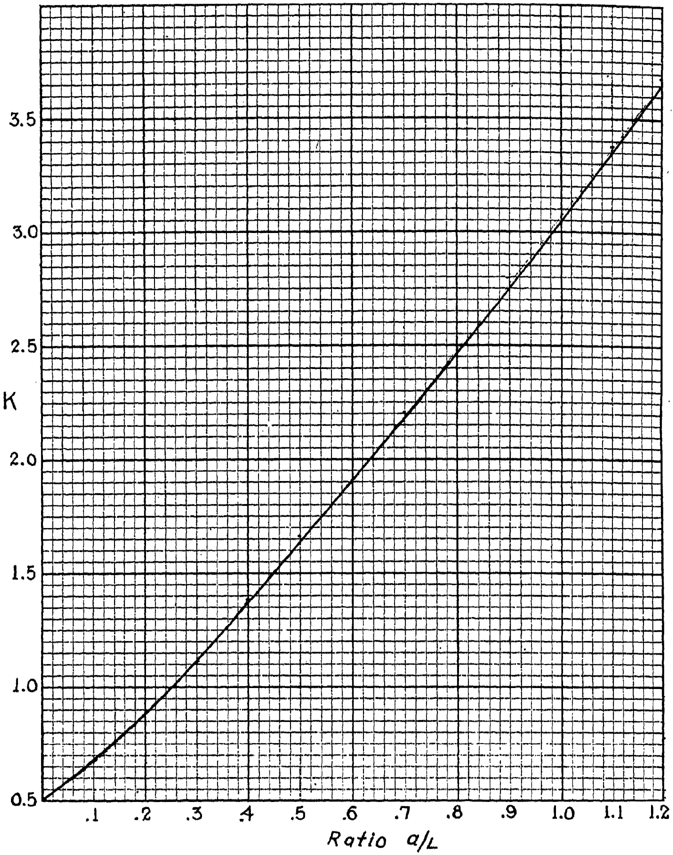


Fig. 178. Chart for Determining Coefficient "K"

Based on this allowable unit stress, the following values have been computed for various sizes of fillets (leg dimension) (see Table 45).

Table 45. - Allowable Loads in Lug Welds, Pounds per Linear Inch

Size of Fillet	Load	Size of Fillet	Load
$\frac{1}{4}$	950	$\frac{1}{2}$	1,900
$\frac{5}{16}$	1,180	$\frac{9}{16}$	2,140
$\frac{3}{8}$	1,420	$\frac{5}{8}$	2,370
$\frac{7}{16}$	1,660	$\frac{3}{4}$	2,850

Anchor Bolts for Lug Supports. - The anchor bolts for lug supports are calculated in the same way as described for leg supports. In calculating the wind load, the exposed surface above the level of support only is taken into consideration.

Skirt Supports. - Skirt supports are located preferably in line with the centerline of the shell (Fig. 179), or attached to the shell clear of the head intersection (Fig. 180). If a large diameter of the bolt circle is required on account of a high wind load, a conical skirt (Fig. 181) is employed. In some cases the shell of the drum is extended and serves as the skirt, the bottom being attached by a lap joint (Fig. 182). However, this type of support is, in general, less satisfactory. In order to facilitate the installation of a tower, it is advisable to provide pipe sleeves through the anchor bolt holes in the base ring and in the upper plate so that the anchor bolts can slip easily through the two holes.

Design of Skirt Weld. - The welded joint between skirt and drum is subjected to shear, and the highest stresses occur when the drum is full and subjected to wind load. The maximum stress, in pounds per lineal inch of circumferential weld, is

$$S_{max} = \frac{0.0533\ p \cdot h^2 \cdot D}{d^2} + \frac{W}{12\pi d} \qquad (20)$$

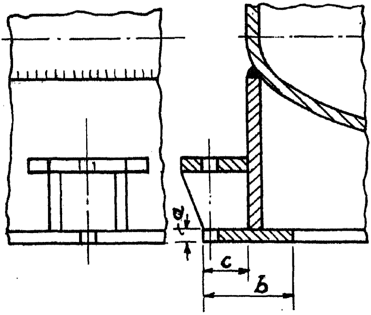


Fig. 179

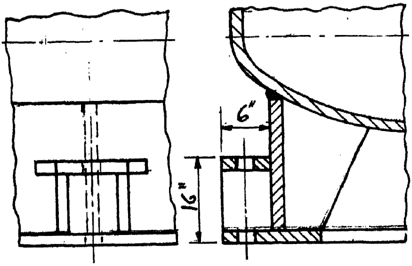


Fig. 180

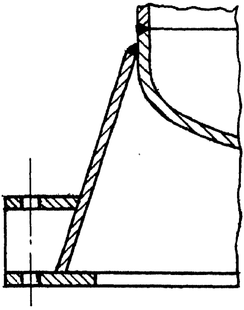


Fig. 181

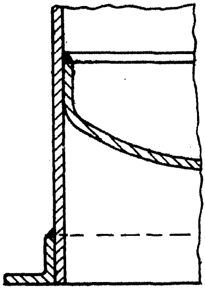


Fig. 182

where p = unit wind pressure, lb. per sq. ft.
h = height of drum above skirt weld, ft.
D = diameter of exposed area, ft.
d = diameter of skirt weld, ft.
W = weight of full drum, lb.

The throat dimension of the weld in inches is

$$t_1 = S_{max}/E \cdot s. \tag{21}$$

where s = allowable working stress of weld, lb. per sq. in.
E = efficiency of joint

The joint efficiency of a single full-fillet lap joint under shear is .55 x .8 = 44%, and of a full-fillet lap joint with plug welds .65 x .8 = .52%. For stress relieved vessels both factors are multiplied by 1.06. The size of the weld determines the thickness of the skirt plate.

Anchor Bolts of Skirt Supports. - For self-supporting stacks, and towers supported by skirts, the stresses in anchor bolts are usually based on the stress per lineal inch of circumference. In accordance with equation (5), Chapter XII, the stress per lineal inch of bolt circle circumference due to wind load is

$$S_1 = \frac{0.0533\, p \cdot h^2 \cdot D}{d_1^{\,2}} \tag{22}$$

where D = diameter of exposed surface, ft.
d_1 = diameter of bolt circle, ft.
h = total height of tower, ft.

From this equation the load, in pounds, for one anchor bolt due to wind is

$$P = \frac{0.0533\, g \cdot p \cdot h^2 \cdot D}{d_1^{\,2}} \tag{23}$$

where g = spacing of the anchor bolts, in.

The dead load of the shell per lineal inch of circumference has to be deducted from the wind load, as above, in order to obtain the net tension.

$$P_1 = \frac{0.0533\ g \cdot p \cdot h^2 \cdot D}{d_1^2} - \frac{g \cdot W_1}{12\pi d_1} \qquad (24)$$

where W_1 = weight of empty drum, lb.

For p = 25 lb. per sq. ft.

$$P_1 = \frac{1.33\ g \cdot h^2 \cdot D}{d_1^2} - \frac{g \cdot W_1}{12\pi d_1} \qquad (25)$$

The bolt area is then computed as above.

For self-supporting steel stacks, the anchor bolts are made not less than 1-1½ in. in diameter, and preferably arranged in pairs, one on the inside and the other on the outside of the steel shell. The anchor bolt circle of high towers is usually considerably larger than the diameter of the shell in order to provide for a sufficient number of anchor bolts.

Base Plate of Skirt Support. - The maximum compression on the base plate per lineal inch of the anchor bolt circle from equations (20) and (22) is

$$F' = \frac{0.0533\ p \cdot h^2 \cdot D}{d_1^2} + \frac{W_1 + W_2}{12\pi d_1} \qquad (26)$$

where W_1 = weight of drum, lb.
W_2 = weight of liquid, lb.

By treating the base plate as a cantilever beam with thickness a and span c (Fig. 179), the thickness of the base, in inches, is computed

$$a = c\sqrt{3W/s} \qquad (27)$$

where a = thickness of base plate, in.
b = width of base plate, in.
c = projection of base plate, in.
$W = F'/b$ = pressure of the steel base plate on the masonry, lb. per sq. in.
s = allowable tensile stress in steel plate, lb. per sq. in.

REFERENCES

1. M. S. Ketchum, Structural Engineer's Handbook, New York, 1924.
2. American Institute of Steel Construction, Steel Construction, 1934 edition, New York.
3. American Welding Society, Welding Handbook, 1938 edition.
4. The Welding Engineer Publishing Co., The Welding Encyclopedia, Chicago, 1938.

APPENDIX

Table A-1.--Codes Regulating the Construction of Pressure Vessels

Boiler Code Committee of the American Society of Mechanical Engineers, 29 West 39th Street, New York, N.Y.,

Boiler Code

Section	I	Power Boilers,
"	II	Materials,
"	III	Boilers of Locomotives,
"	IV	Low Pressure Heating Boilers,
"	V	Miniature Boilers,
"	VI	Rules for Inspection,
"	VII	Suggested Rules for Care of Power Boilers,
"	VIII	Unfired Pressure Vessels.

Joint API-ASME Committee on Unfired Pressure Vessels;

API-ASME Code - Unfired Pressure Vessels for Petroleum Liquids and Gases.

State and City Codes or Regulations.

Bureau of Marine Inspection and Navigation.

Bureau of Engineering, U. S. Navy.

American Bureau of Shipping.

Lloyd's Register of Shipping.

Heat Exchange Institute, 90 West Street, New York, N.Y.

"Synopsis of boiler laws, rules and regulations by states, provinces and cities (U. S. and Canada), Jan. 1, 1939," published by the National Bureau of Casualty and Surety Underwriters, Boiler and Machinery Division, 60 John St., New York, N. Y.

Table A-2.--ALLOWABLE WORKING STRESSES FOR FERROUS MATERIALS, IN LB PER SQ IN.--Factor of Safety = 5 (A.S.M.E. Code)

Specification number	Grade	Spec., minimum tensile	Notes and limitations	For metal temperatures not exceeding deg F —20 to 650	700	750	800	850	900	950	1000	1050	1100	1150	1200
PLATE STEELS															
Carbon Steel															
S-1		55000	(1)	11000	10400	9500	8000	6300	4400	2600	..	..	..	..	..
S-2	A	45000	(1)	9000	8800	8400	6900	5700	4400	2600	..	..	..	..	..
S-2	B	50000	(1)	10000	9600	9000	7500	6000	4400	2600	..	..	..	..	..
S-42	A	55000		11000	10400	9500	8500	7200	5600	3800	2000	..	..	..	..
S-42	B	60000		12000	11400	10400	9100	7400	5600	3800	2000	..	..	..	..
S-55	A	65000		13000	12300	11100	9400	7600	5600	3800	2000	..	..	..	..
S-55	B	70000		14000	13300	11900	10000	7800	5600	3800	2000	..	..	..	..
Low-Alloy Steels															
S-28	A	75000		15000	14100	12400	10100	7800	5600	3800	2000	..	..	..	..
S-28	B	85000		15000	14100	12400	10100	7800	5600	3800	2000	..	..	..	..
S-43	A	65000		13000	12300	11100	9400	7600	5600	3800	2000	..	..	..	..
S-43	B	70000		14000	13300	11900	10000	7800	5600	3800	2000	..	..	..	..
S-43	C	75000		15000	14100	12400	10100	7800	5600	3800	2000	..	..	..	..
S-44	A	65000		13000	13000	13000	12500	11500	10000	8000	5000	..	..	..	..
S-44	B	70000		14000	14000	14000	13500	12000	10200	8000	5000	..	..	..	..
S-44	C	75000		15000	15000	15000	14400	12700	10400	8000	5000	..	..	..	..
Medium and High Alloy Steels															
A.S.T.M. A176-39	1	70000		14000	14000	14000	12800	9500	6750	4000	2400	..	..	..	..
A.S.T.M. A176-39	2	70000		14000	14000	14000	12800	9500	6750	4000	2400	..	..	..	..
A.S.T.M. A167-39	304, 308	75000	(2)	15000	15000	14000	13000	12000	11000	10500	10000	8500	5600	4200	3200
A.S.T.M. A167-39	316, 317	75000	(2)	15000	15000	14000	14000	13900	13400	12600	11200	9000	7000	5000	3600
A.S.T.M. A167-39	321	75000		15000	15000	14000	13000	12000	11000	10500	10000	8500	5600	4200	3200
A.S.T.M. A167-39	347	75000		15000	15000	14000	13000	12000	11000	10500	10000	8500	5600	4200	3200
FORGINGS															
Carbon Steels															
S-4	1	60000		12000	11400	10400	9100	7400	5600	3800	2000	..	..	..	..
S-4	2	70000		14000	13300	11900	10000	7800	5600	3800	2000	..	..	..	..
S-4	3	75000		15000	14100	12400	10100	7800	5600	3800	2000	..	..	..	..
S-8	1	60000		12000	11400	10400	8300	6350	4400	2600	..	..	..	..	..
S-8	2	70000		14000	13300	11900	8950	6450	4400	2600	..	..	..	..	..
S-8	1	60000	(3)	12000	11400	10400	9100	7400	5600	3800	2000	..	..	..	..
S-8	2	70000	(3)	14000	13300	11900	10000	7800	5600	3800	2000	..	..	..	..
S-10	A-Untreated	47000		9400	9100	8600	7150	5850	4400	2600	..	..	..	..	..
S-10	B-Untreated	60000		12000	11400	10400	8300	6350	4400	2600	..	..	..	..	..
S-10	C-Annealed	60000		12000	11400	10400	8300	6350	4400	2600	..	..	..	..	..
S-50	1	60000		12000	11400	10400	8300	6350	4400	2600	..	..	..	..	..
S-50	2	70000		14000	13300	11900	8950	6450	4400	2600	..	..	..	..	..
S-50	1	60000	(3)	12000	11400	10400	9100	7400	5600	3800	2000	..	..	..	..
S-50	2	70000	(3)	14000	13300	11900	10000	7800	5600	3800	2000	..	..	..	..
Alloy Steels															
S-35	F1	70000		14000	14000	14000	13500	12000	10200	8000	5000	..	..	..	..
S-35	F3	70000		14000	14000	14000	14000	13400	11000	8250	5850	3850	2200	..	..
S-35	F4	90000	(13)	15000	15000	15000	14000	12500	10000	..	..	..	..	..	..
S-35	F5	90000	(7)	15000	15000	15000	14000	13400	11000	8250	5850	3850	2200	..	..
S-35	F6	85000		15000	14000	13000	11500	9500	6750	4000	2400	..	..	..	..
S-35	F7	100000		15000	15000	15000	14400	12700	10400	8000	5000	..	..	..	..
S-35	F8	75000	(2)	15000	15000	14000	13000	12000	11000	10500	10000	8500	5600	4200	3200
S-35	F11	100000		15000	15000	15000	14400	12700	10400	8000	5000	..	..	..	..
BOLTING															
Carbon Steel															
S-7		55000	(10)	...	...	...	...	...	...	...	...	..	..	..	..
Alloy Steel															
S-9	A		(11), (12)	13000	11950	10900	...	...	...	...	...	..	..	..	..
S-9	B		(11), (12)	15000	13750	12500	...	...	...	...	...	..	..	..	..
S-9	C		(11), (12)	16000	14700	13400	...	...	...	...	...	..	..	..	..
A.S.T.M. A193-39T	B4		(11), (12)	16000	16000	16000	16000	13000	10000	...	...	..	..	..	..
A.S.T.M. A193-39T	B5		(7), (11), (12)	16000	16000	16000	16000	13800	11000	8250	5850	3850	2200	..	..
A.S.T.M. A193-39T	B6		(11), (12)	16000	14700	13400	11500	9500	6750	...	...	..	..	..	..
A.S.T.M. A193-39T	B7		(11), (12)	16000	16000	16000	16000	13000	10000	...	...	..	..	..	..
A.S.T.M. A193-39T	B7a		(11), (12)	16000	16000	16000	16000	13800	11000	8250	5850	3850	2200	..	..
A.S.T.M. A193-39T	B8		(2), (11)	15000	15000	14000	13000	12000	11000	10500	10000	8500	5000	4200	3200
A.S.T.M. A193-39T	B13		(11), (12)	16000	16000	16000	16000	13800	11000	8250	5850	3850	2200	..	..
A.S.T.M. A193-39T	B14		(11), (12)	16000	16000	16000	16000	15000	13300	11400	8800	5000	2200	..	..
A.S.T.M. A1															

NOTES:

(1) Flange quality not permitted above 850 F.

(2) No allowance has been made for corrosive action in the allowable stresses given. Carbide precipitation in service is also to be expected at temperatures above 750 F.

(3) These stresses permitted only if 0.10 per cent minimum silicon is expressly specified.

(7) These stresses permitted only with molybdenum.

(11) These stresses are established from a consideration of strength only and will be satisfactory for average service. For bolted joints where freedom from leakage over a long period of time without retightening is required, lower stresses may be necessary as determined from the relative flexibility of the flange and bolts and corresponding relaxation properties.

(12) Between temperatures of —20 to 400 F, stresses equal to the lower of the following will be permitted. 16 per cent of tensile strength; 20 per cent of yield point stress.

(13) These materials may be used at higher temperatures with suitable stress allowances. Agreement has not been reached as to what these values should be.

GENERAL:

Carbon steel materials covered by Boiler Code specifications which are not included in these tables are limited to use at temperatures of from —20 to 650 F.

The allowable stress in wrought materials is one-fifth of the ultimate; in cast materials, it is one-fifth of the ultimate multiplied by the applicable quality factor for design purposes.

Table A-3.--Allowable Working Stresses for Non-ferrous Materials,
Lb. Per Sq. In.
(A.S.M.E. Code)

Material	Spec. Number	For temperatures not exceeding deg. F.								
		Sub zero	70 to 100	150	250	350	406	450	500	550
Muntz metal tubing and high brass tubing	S-24	5,000	5,000	5,000	4,000	2,500	.....	.....	.....	......
Muntz metal condenser tubes	S-47	5,000	5,000	5,000	4,000	2,500	.....	.....	.....	......
Red brass tubes	S-24	6,000	6,000	6,000	5,500	5,000	4,500	.....	.....	......
Copper plates	S-20	6,000	6,000	6,000	5,000	4,500	4,000	.....	.....	......
Copper tubes	S-22	6,000	6,000	6,000	5,000	4,500	4,000	.....	.....	......
Copper pipe	S-23	6,000	6,000	6,000	5,000	4,500	4,000	.....	.....	......
Admiralty tubes	S-24	7,000	7,000	7,000	6,500	6,000	5,500	4,500	......	......
Admiralty condenser tubes	S-47	7,000	7,000	7,000	6,500	6,000	5,500	4,500	.....	......
Copper silicon alloy plates, rods, bars, and shapes	S-36 S-37(1)	10,000	10,000	10,000	10,000	5,000	......	......	.....	......
Monel	(2)	14,000	14,000	14,000	14,000	14,000	14,000	13,500	13,000	12,500
Steam bronze	S-41	6,800	6,800	6,800	6,300	5,800	5,400	5,000	4,200	3,300
Steam bronze	S-46(3)	5,500	5,500	5,500	5,000	4,500	3,500	......	......	....
Cupro nickel	S-47(4)	10,000	10,000	10,000	10,000	9,000	8,400	7,500	6,300	5,000

(1) Type A and C only.

(2) Applies to 70,000 lb. tensile strength rolled and annealed material.

(3) A.S.T.M. Specification B62-36.

(4) A.S.T.M. Specification B111-37T, 70-30 Copper-Nickel and 80-20 Copper-Nickel Type A and B only.

Table A-4.--Cylindrical Shells--Internal Pressure--Carbon Steel
Allowable Working Pressures, Lb. Per Sq. In.

Joint Efficiency--100%; Factor of Safety--5; Tensile Strength--55,000.

Inside Diameter Inches	Thickness--Inches														
	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$
24	229	286	354	401	458	516	573	630	687	745	802	859	917	1031	1039*)
26	212	264	317	370	423	480	529	582	534	687	740	793	846	952	1056
28	196	246	294	344	393	442	491	540	589	638	687	737	786	884	982
30	183	229	275	320	367	412	458	504	550	596	641	688	733	825	916
32	172	215	258	300	344	387	430	475	515	558	601	646	687	773	846
34	162	202	243	283	324	364	404	445	485	526	566	606	647	728	809
36	153	191	229	267	306	344	382	420	458	496	535	573	611	688	764
38	145	181	217	253	289	325	362	398	434	470	506	543	579	651	724
40	137	172	206	240	275	309	344	378	412	447	481	516	550	619	687
42	131	164	196	229	262	294	327	360	393	425	458	491	524	589	655
48	115	143	172	219	229	258	286	315	344	372	400	429	458	516	573
54	102	127	153	209	204	229	254	280	305	331	356	382	408	458	509
60	92	115	137	200	183	206	229	252	275	298	321	344	367	413	458
66	83	104	125	146	167	187	208	229	250	271	292	313	333	375	417
72	76	95	115	134	153	172	191	210	229	248	267	286	306	344	382
78	71	88	106	123	140	158	176	194	212	229	247	264	282	317	353
84	65	82	98	114	131	147	163	180	196	213	229	246	262	295	327
90	61	76	92	107	122	137	153	168	183	198	214	229	249	275	306
96	57	72	86	100	115	129	143	158	172	186	201	215	229	258	286
102	54	67	81	94	108	121	135	148	162	175	189	202	216	243	270
108	51	64	76	89	102	114	127	140	153	166	178	191	204	229	255
114	48	60	72	84	97	108	120	132	145	157	169	181	193	217	241
120	46	57	69	80	92	103	114	126	137	149	160	172	183	206	229

*)Based on outside diameter formula.

Table A-5.--Cylindrical Shells--Internal Pressure-Stainless Steel
Allowable Working Pressures, Lb. Per Sq. In.

Joint Efficiency--100%; Factor of Safety--5;
Tensile Strength--80,000 Lb. Per Sq. In.

Inside Diameter, Inches	Thickness--Inches								
	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$
12	333	500	666	683	1,000	1,163	1,332	1,500	1,509*
18	222	333	445	555	666	777	888	999	1,110
24	166	250	333	416	500	582	666	750	833
30	133	200	267	333	400	467	534	600	666
36	111	166	222	277	333	388	444	500	555
42	95	143	190	238	286	333	381	429	477
48	83	125	166	208	250	292	333	375	416
54	74	111	148	185	222	259	296	333	368
60	67	100	133	166	200	233	266	300	333
66	61	91	121	151	182	212	242	273	303
72	55	83	111	138	166	194	222	250	278
78	51	77	103	128	154	179	205	231	256
84	48	71	95	118	132	166	190	214	238
90	45	67	89	111	133	156	178	200	222
96	42	63	83	104	125	146	166	187	208

*Based on outside diameter formula.

Table A-6. Cylindrical Shells--Internal Pressure--Monel
Allowable Working Pressures, Lb. Per Sq. In.

Joint Efficiency--100%; Factor of Safety--5;
Tensile Strength--70,000 Lb. Per Sq. In.

Inside Diameter, Inches	Thickness--Inches								
	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$
12	292	438	584	730	875	1,020	1,168	1,312	1,320*
18	195	292	390	486	585	681	780	875	972
24	146	219	292	364	438	510	583	657	730
30	117	175	234	292	350	419	468	525	584
36	97	146	195	243	292	340	389	438	486
42	83	125	167	208	250	292	333	375	417
48	73	109	146	182	219	256	292	328	364
54	65	92	130	162	195	227	259	292	322
60	58	87	117	146	175	204	234	263	292
66	53	80	106	133	159	185	212	239	265
72	49	73	97	122	146	170	195	219	243
78	45	67	90	112	135	157	180	202	224
84	42	62	83	104	125	146	167	188	208
90	39	58	78	97	117	136	156	175	195
96	36	55	73	91	109	128	146	164	182

*Based on outside diameter formula.

Table A-7.--Cylindrical Shells--Internal Pressure--Copper
Allowable Working Pressure, Lb. Per Sq. In.

Joint Efficiency--100%; Factor of Safety--5;
Tensile Strength--30,000 Lb. Per Sq. In.

Inside Diameter, Inches	Thickness--Inches										
	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$
12	125	187	250	312	375	437	500	563	566	581	629
18	83	125	167	208	250	292	333	375	416	459	500
24	62	94	125	156	188	218	250	282	313	344	375
30	50	75	100	125	150	175	200	225	250	275	300
36	42	62	83	104	125	146	167	188	208	229	250
42	36	53	71	89	107	125	143	161	179	197	214
48	31	47	62	78	94	109	125	141	157	172	188
54	28	42	56	69	83	97	111	125	138	153	167
60	25	37	50	62	75	87	100	112	125	137	150
66	23	34	45	57	68	79	91	102	114	125	136
72	21	31	41	52	62	73	83	94	104	114	125
78	19	29	38	48	58	67	77	86	96	105	115
84	18	26	35	44	53	62	71	80	89	98	107
90	17	25	33	42	50	58	67	75	83	92	100
96	16	23	31	39	47	54	62	70	78	86	94

Underlined values based on outside diameter formula.

Table A-8.--Cylindrical Shells--Internal Pressure--Aluminum Allowable Working Pressures, Lb. Per Sq. In.

Joint Efficiency--100%; Factor of Safety--5;
Tensile Strength--13,000 Lb. Per Sq. In.;
Yield Strength--[0.2% set]--4,000 Lb. Per Sq. In.

Inside Diameter, Inches	Thickness--Inches								
	1/4	3/8	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4
12	108	162	217	246	289	331	372	411	450
18	72	108	145	180	217	253	260	289	318
24	54	82	108	135	163	189	217	244	246
30	43	65	87	108	130	152	173	195	217
36	36	54	72	90	108	127	145	163	180
42	31	46	62	77	93	108	124	140	155
48	27	41	54	68	81	94	108	122	136
54	24	36	49	60	72	84	96	108	120
60	22	32	43	54	65	75	87	97	108
66	20	29	39	49	59	68	79	88	99
72	18	27	36	45	54	63	72	81	90
78	17	25	33	42	50	58	67	75	83
84	16	23	30	38	46	54	61	69	77
90	15	22	29	36	43	50	58	65	72
96	14	20	27	34	41	47	54	61	68

Values above heavy line are based on outside diameter formula.

Table A-9. - Cylindrical Shells--Internal Pressure--Carbon Steel

Allowable Working Pressure, lb. per sq. in.
Riveted Butt Joints

(The Hartford Steam Boiler Inspection and Insurance Co.)

Plate Thickness Inches	Diameter of Shell--Inches									
	24	30	36		42		48		54	
	Double Riveted	Double Riveted	Double Riveted	Triple Riveted	Double Riveted	Triple Riveted	Double Riveted	Triple Riveted	Double Riveted	Triple Riveted
$\frac{1}{4}$	189.7	151.8	126.5	133.6	108.4	114.5	94.8	100.2	84.3	89.1
$\frac{9}{32}$	213.4	170.7	142.3	150.3	121.9	128.9	106.7	112.7	94.8	100.2
$\frac{5}{16}$	234.6	187.6	156.4	167.1	134.0	143.2	117.3	125.3	104.2	111.4
$\frac{11}{32}$	258.0	206.4	172.0	183.8	147.4	157.5	129.0	137.8	114.6	122.5
$\frac{3}{8}$	281.5	225.2	187.6	202.5	160.8	173.6	140.7	151.9	125.1	135.0
$\frac{13}{32}$	304.9	243.9	203.3	219.4	174.2	188.1	152.4	164.5	135.5	146.2
$\frac{7}{16}$	326.0	260.8	217.3	235.0	186.3	201.4	163.0	176.2	144.9	156.6
$\frac{15}{32}$	349.3	279.4	232.8	251.7	199.6	215.8	174.6	188.8	155.2	167.8
$\frac{1}{2}$	372.6	298.1	248.4	269.8	212.9	231.2	186.3	202.3	165.6	179.8
$\frac{17}{32}$				286.6		245.7		215.0		191.1
$\frac{9}{16}$				298.0		255.4		223.5		198.6
$\frac{19}{32}$				314.5		269.6		235.9		209.7
$\frac{5}{8}$				331.1		283.8		248.3		220.7
$\frac{21}{32}$				347.7		298.0		260.7		231.8
$\frac{11}{16}$				359.6		308.2		269.7		239.7
$\frac{23}{32}$				375.9		322.2		281.9		250.6
$\frac{3}{4}$				391.8		335.8		293.9		261.2
$\frac{25}{32}$				403.9		346.2		302.9		269.2
$\frac{13}{16}$				420.0		360.0		315.0		280.0
$\frac{27}{32}$				434.1		372.1		325.5		289.4
$\frac{7}{8}$				449.7		385.4		337.2		299.8
$\frac{29}{32}$				462.9		396.8		347.2		308.6
$\frac{15}{16}$				479.5		411.0		359.6		319.6
$\frac{31}{32}$				492.5		422.1		369.4		328.3
1				509.6		436.8		382.2		339.7

Table A-9 (Cont'd). - Cylindrical Shells--Internal Pressure--Carbon Steel

Allowable Working Pressures, lb. per sq. in.
Riveted Butt Joints

(The Hartford Steam Boiler Inspection and Insurance Co.)

Plate Thickness Inches	Diameter of Shell--Inches								
	60			66			72		
	Double Riveted	Triple Riveted	Quadruple Riveted	Double Riveted	Triple Riveted	Quadruple Riveted	Double Riveted	Triple Riveted	Quadruple Riveted
$\frac{1}{4}$	75.8	80.2	85.9	68.9	72.9	78.1	63.2	66.8	71.6
$\frac{9}{32}$	85.3	90.2	96.7	77.6	82.0	87.9	71.1	75.1	80.6
$\frac{5}{16}$	93.8	100.2	107.4	85.3	91.1	97.7	78.2	83.5	89.5
$\frac{11}{32}$	103.2	110.2	118.2	93.8	100.2	107.4	86.0	91.9	98.5
$\frac{3}{8}$	112.6	121.5	129.5	102.3	110.5	117.7	93.8	101.2	107.9
$\frac{13}{32}$	121.9	131.6	140.3	110.9	119.7	127.5	101.6	109.7	116.9
$\frac{7}{16}$	130.4	141.0	150.7	118.5	128.1	137.0	108.6	117.5	125.5
$\frac{15}{32}$	139.7	151.0	161.5	127.0	137.3	146.8	116.4	125.8	134.6
$\frac{1}{2}$	149.0	161.8	172.5	135.5	147.1	156.8	124.2	134.9	143.7
$\frac{17}{32}$		172.0	183.2		156.3	166.6		143.3	152.7
$\frac{9}{16}$		178.8	192.6		162.5	175.1		149.0	160.5
$\frac{19}{32}$		188.7	203.3		171.6	184.8		157.2	169.4
$\frac{5}{8}$		198.6	214.0		180.6	194.5		165.5	178.3
$\frac{21}{32}$		208.6	224.7		189.6	204.3		173.8	187.2
$\frac{11}{16}$		215.7	233.9		196.1	212.6		179.8	194.9
$\frac{23}{32}$		225.5	244.5		205.0	222.3		187.9	203.8
$\frac{3}{4}$		235.1	254.9		213.7	231.7		195.9	212.4
$\frac{25}{32}$		242.3	264.3		220.3	240.3		201.9	220.3
$\frac{13}{16}$		252.0	274.9		229.1	249.9		210.0	229.1
$\frac{27}{32}$		260.4	284.0		236.7	258.1		217.0	236.6
$\frac{7}{8}$		269.8	292.6		245.2	266.0		224.8	243.8
$\frac{29}{32}$		277.7	300.7		252.5	273.3		231.4	250.6
$\frac{15}{16}$		287.7	309.7		261.5	281.5		239.7	258.0
$\frac{31}{32}$		295.5	317.9		268.6	289.0		246.2	264.9
1		305.8	330.7		278.0	300.6		254.8	275.6
$1\frac{1}{32}$		338.8	338.8			308.0			282.3
$1\frac{1}{16}$		346.7	346.7			315.2			288.9
$1\frac{3}{32}$		354.9	354.9			322.6			295.7
$1\frac{1}{8}$		363.0	363.0			330.0			302.5
$1\frac{5}{32}$		370.9	370.9			337.2			309.1
$1\frac{3}{16}$		381.8	381.8			347.1			318.2
$1\frac{7}{32}$		389.6	389.6			354.2			324.7
$1\frac{1}{4}$		397.8	397.8			361.6			331.5

Table A-9 (Cont'd) - Cylindrical Shells--Internal Pressure--Carbon Steel

Allowable Working Pressures, lb. per sq. in.
Riveted Butt Joints

(The Hartford Steam Boiler Inspection and Insurance Co.)

Plate Thickness Inches	Diameter of Shell--Inches							
	78		84		90	96	102	108
	Triple Riveted	Quadruple Riveted	Triple Riveted	Quadruple Riveted	Quadruple Riveted	Quadruple Riveted	Quadruple Riveted	Quadruple Riveted
$\frac{1}{4}$	61.6	66.1	57.2	61.4	57.3	53.7	50.5	47.7
$\frac{9}{32}$	69.4	74.4	64.4	69.0	64.4	60.4	56.8	53.7
$\frac{5}{16}$	77.1	82.6	71.6	76.7	71.6	67.1	63.2	59.7
$\frac{11}{32}$	84.8	90.9	78.7	84.4	78.8	73.8	69.5	65.6
$\frac{3}{8}$	93.5	99.6	86.8	92.5	86.3	80.9	76.1	71.9
$\frac{13}{32}$	101.2	107.9	94.0	100.2	93.5	87.6	82.5	77.9
$\frac{7}{16}$	108.4	115.9	100.7	107.7	100.5	94.2	88.7	83.7
$\frac{15}{32}$	116.2	124.2	107.9	115.3	107.7	100.9	95.0	89.7
$\frac{1}{2}$	124.5	132.7	115.6	123.2	115.0	107.8	101.4	95.8
$\frac{17}{32}$	132.2	140.9	122.8	130.9	122.1	114.5	107.8	101.8
$\frac{9}{16}$	137.5	148.1	127.7	137.5	128.4	120.3	113.3	107.0
$\frac{19}{32}$	145.1	156.4	134.8	145.2	135.6	127.1	119.6	112.9
$\frac{5}{8}$	152.8	164.6	141.9	152.8	142.6	133.7	125.9	118.9
$\frac{21}{32}$	160.4	172.8	149.0	160.5	149.8	140.4	132.2	124.8
$\frac{11}{16}$	165.9	179.9	154.1	167.0	155.9	146.2	137.6	129.9
$\frac{23}{32}$	173.4	188.1	161.1	174.6	163.0	152.8	143.8	135.8
$\frac{3}{4}$	180.8	196.0	167.9	182.0	169.9	159.3	149.9	141.6
$\frac{25}{32}$	186.4	203.3	173.1	188.8	176.2	165.2	155.5	146.8
$\frac{13}{16}$	193.8	211.5	180.0	196.4	183.3	171.8	161.7	152.7
$\frac{27}{32}$	200.3	218.4	186.0	202.8	189.3	177.5	167.1	157.7
$\frac{7}{8}$	207.5	225.0	192.7	209.0	195.0	182.8	172.1	162.5
$\frac{29}{32}$	213.6	231.3	198.4	214.8	200.4	187.9	176.8	167.0
$\frac{15}{16}$	221.3	238.2	205.5	221.2	206.4	193.5	182.1	172.0
$\frac{31}{32}$	227.3	244.5	211.0	227.0	211.9	198.6	187.0	176.6
1	235.2	254.4	218.4	236.2	220.4	206.7	194.5	183.7
$1\frac{1}{32}$		260.6		242.0	225.8	211.7	199.2	188.2
$1\frac{1}{16}$		266.7		247.6	231.1	216.7	203.9	192.6
$1\frac{3}{32}$		273.0		253.5	236.6	221.8	208.7	197.1
$1\frac{1}{8}$		279.2		259.2	242.0	226.8	213.5	201.6
$1\frac{5}{32}$		285.3		264.9	247.3	231.8	218.2	206.0
$1\frac{3}{16}$		293.7		272.7	254.5	238.6	224.6	212.1
$1\frac{7}{32}$		299.7		278.3	259.7	243.5	229.2	216.4
$1\frac{1}{4}$		306.0		284.1	265.2	248.6	234.0	221.0

Table A-10. - Spherical Dished Heads--Pressure on Concave Side--Carbon Steel

Allowable Working Pressures, lb. per sq. in.
A.S.M.E. Code, 1937 edition
Tensile Strength--55,000 lb. per sq. in.

Thickness, Inches	Radius to Which Head is Dished, Inches									
	12	14	16	18	20	22	24	30	36	42
$\frac{1}{4}$"	275.1	235.8	206.3	183.4	165.0	150.0	137.6	110.0	91.7	78.6
$\frac{5}{16}$"	343.8	294.8	257.9	229.2	206.3	187.6	171.9	137.5	114.6	98.3
$\frac{3}{8}$"	412.7	353.7	309.5	275.1	247.5	225.1	206.3	165.1	137.6	117.9
$\frac{7}{16}$"	481.4	412.6	361.1	320.9	288.8	262.6	240.7	192.6	160.5	137.6
$\frac{1}{2}$"	550.2	471.6	412.6	366.8	330.1	300.1	275.1	220.1	183.4	157.2
$\frac{9}{16}$"	619.0	530.5	464.2	412.6	371.4	337.6	309.5	247.6	206.3	176.9
$\frac{5}{8}$"	687.8	589.5	515.8	458.5	412.6	375.1	343.9	275.1	229.3	196.5
$\frac{11}{16}$"	756.5	648.4	567.4	504.3	453.9	412.6	378.3	302.6	252.2	216.2
$\frac{3}{4}$"	825.3	707.4	618.9	550.2	495.1	450.1	412.7	330.1	275.1	235.8
$\frac{13}{16}$"	894.0	766.4	670.5	596.0	536.4	487.7	447.0	357.6	298.0	255.4
$\frac{7}{8}$"	962.8	825.3	722.1	641.9	577.7	525.2	481.4	385.1	320.9	275.1
$\frac{15}{16}$"	1,031.6	884.3	773.7	687.7	618.9	562.7	515.8	412.6	343.8	294.7
1"	1,100.4	943.2	825.3	733.6	660.2	600.2	550.2	440.1	366.8	314.5
$1\frac{1}{16}$"	1,169.1	1,002.1	876.8	779.4	701.5	637.7	584.5	467.6	389.7	334.0
$1\frac{1}{8}$"	1,237.0	1,061.2	928.4	825.2	742.7	675.2	618.9	495.1	412.6	353.7
$1\frac{3}{16}$"	1,306.6	1,120.0	980.0	871.1	784.1	712.7	653.3	522.6	435.5	373.3
$1\frac{1}{4}$"	1,375.5	1,179.0	1,031.6	917.0	825.3	750.2	687.7	550.2	458.5	393.0
$1\frac{3}{8}$"	1,513.0	1,296.9	1,134.8	1,008.7	907.8	825.3	756.5	605.2	504.4	432.3
$1\frac{1}{2}$"	1,650.5	1,414.8	1,237.9	1,100.4	990.3	900.3	825.3	660.2	550.2	471.6

Table A-10 (Cont'd). - Spherical Dished Heads--Pressure on Concave Side--Carbon Steel

Allowable Working Pressures, lb. per sq. in.
A.S.M.E. Code, 1937 edition
Tensile Strength--55,000 lb. per sq. in.

Thickness, Inches	Radius to Which Head is Dished, Inches												
	48	54	60	66	72	78	84	90	96	102	108	114	120
$\frac{1}{4}$"	68.8	61.1	55.1	50.0	45.9	42.3	39.3	36.7	34.4	32.4	30.6	29.0	27.5
$\frac{5}{16}$"	86.0	76.4	68.8	62.5	57.3	52.9	49.1	45.9	43.0	40.5	38.2	36.2	34.4
$\frac{3}{8}$"	103.2	91.7	82.5	75.0	68.8	63.5	59.0	55.0	51.6	48.5	45.9	43.4	41.3
$\frac{7}{16}$"	120.4	107.0	96.3	87.5	80.2	74.1	68.8	64.2	60.2	56.6	53.5	50.7	48.1
$\frac{1}{2}$"	137.5	122.3	110.0	100.0	91.7	84.6	78.6	73.4	68.7	64.7	61.1	57.9	55.0
$\frac{9}{16}$"	154.7	137.5	123.8	112.5	103.1	95.2	88.4	82.5	77.4	72.8	68.8	65.2	61.9
$\frac{5}{8}$"	171.9	152.8	137.5	125.0	114.6	105.8	98.3	91.7	86.0	80.9	76.4	72.4	68.8
$\frac{11}{16}$"	189.1	168.1	151.4	137.6	126.1	116.3	108.0	100.9	94.6	89.0	84.1	79.6	75.7
$\frac{3}{4}$"	206.3	183.4	165.1	150.0	137.6	126.9	117.9	110.1	103.2	97.1	91.7	86.9	82.5
$\frac{13}{16}$"	223.5	198.7	178.8	162.6	149.0	137.6	127.7	119.2	111.8	105.2	99.3	94.1	89.4
$\frac{7}{8}$"	240.7	214.0	192.6	175.1	160.5	148.1	137.5	128.4	120.4	113.3	107.0	101.4	96.3
$\frac{15}{16}$"	266.5	236.9	213.2	193.8	177.7	164.0	152.3	142.2	133.3	125.4	118.4	112.2	106.6
1"	275.1	244.5	220.0	200.0	183.4	169.2	157.3	146.7	137.6	129.5	122.3	115.8	110.0
1$\frac{1}{8}$"	309.5	275.1	247.6	225.1	206.3	190.5	176.8	165.1	154.7	145.6	137.5	130.3	123.8
1$\frac{1}{4}$"	343.8	305.6	275.1	250.0	229.2	211.6	196.5	183.4	171.9	161.8	152.8	144.8	137.6
1$\frac{3}{8}$"	378.2	336.2	302.6	275.1	252.2	232.8	216.1	201.7	189.1	178.0	168.1	159.3	151.3
1$\frac{1}{2}$"	412.6	366.8	330.1	300.1	275.1	253.9	235.8	220.0	206.3	194.1	183.4	173.8	165.1

Allowable Working Pressures, lb. per sq. in.
A.S.M.E. Code, 1937 edition
Tensile Strength--55,000 lb. per sq. in.

Thickness, Inches	Radius to Which Head Is Dished, Inches											
	12	14	16	18	20	22	24	30	36	42	48	54
1/4"	165.1	141.5	123.8	110.0	99.0	90.0	82.6	66.0	55.0	47.2	41.3	36.7
5/16"	206.3	176.9	154.7	137.5	123.8	112.6	103.1	82.5	68.8	59.0	51.6	45.8
3/8"	247.6	212.2	185.7	165.1	148.5	135.1	123.8	99.1	82.6	70.7	61.9	55.0
7/16"	288.8	247.6	216.7	192.5	173.3	157.6	144.4	115.6	96.3	82.6	72.2	64.2
1/2"	330.1	283.0	247.6	220.1	198.1	180.1	165.1	132.1	110.0	94.3	82.5	73.4
9/16"	371.4	318.3	278.5	247.6	222.8	202.6	185.7	148.5	123.8	106.1	92.8	82.5
5/8"	412.7	353.7	309.5	275.1	247.6	225.1	206.3	165.1	137.6	117.9	103.1	91.7
11/16"	453.9	389.0	340.4	302.6	272.3	247.6	227.0	181.6	151.3	129.7	113.5	100.9
3/4"	495.2	424.4	371.3	330.1	297.1	270.1	247.6	198.1	165.1	141.5	123.8	110.0
13/16"	536.5	459.8	402.3	357.6	321.8	292.6	268.6	214.6	178.8	153.2	134.1	119.2
7/8"	577.7	495.2	433.3	385.1	346.6	315.1	288.8	231.1	192.5	165.1	144.4	128.4
15/16"	619.0	530.6	464.2	412.6	371.3	337.6	309.5	247.6	206.3	176.8	154.7	137.5
1"	660.2	565.9	495.2	440.2	396.1	360.1	330.1	264.1	220.1	188.7	165.1	146.7
1 1/8"	742.7	636.7	557.0	495.1	445.6	405.1	371.3	297.1	247.6	212.2	185.7	165.1
1 1/4"	825.3	707.4	619.0	550.2	495.2	450.1	412.6	330.1	275.1	235.8	206.3	183.4
1 3/8"	907.8	778.1	680.9	605.2	544.7	495.2	453.9	363.1	302.6	259.4	226.9	201.7
1 1/2"	990.3	848.9	742.7	660.2	594.2	540.2	495.2	396.1	330.1	283.0	247.6	220.1

Table A-11 (Cont'd). - Spherical Dished Heads--Pressure on Convex Side--Carbon Steel

Allowable Working pressures, lb. per sq. in.
A.S.M.E. Code, 1937 edition
Tensile Strength--55,000 lb. per sq. in.

Thickness, Inches	Radius to Which Head is Dished, Inches										
	60	66	72	78	84	90	96	102	108	114	120
$\frac{1}{4}$	33.1	30.0	27.5	25.4	23.6	22.0	20.6	19.4	18.4	17.4	16.5
$\frac{5}{16}$	41.3	37.5	34.4	31.7	29.5	27.5	25.8	24.3	22.9	21.7	20.6
$\frac{3}{8}$	49.5	45.0	41.3	38.1	35.4	33.0	31.0	29.1	27.5	26.1	24.8
$\frac{7}{16}$	57.8	52.5	48.1	44.5	41.3	38.5	36.1	34.0	32.1	30.4	28.9
$\frac{1}{2}$	66.0	60.0	55.0	50.8	47.2	44.0	41.2	38.8	36.7	34.8	33.0
$\frac{9}{16}$	74.3	67.5	61.9	57.1	53.0	49.5	46.4	43.7	41.3	39.1	37.1
$\frac{5}{8}$	82.5	75.0	68.8	63.5	59.0	55.0	51.6	48.5	45.8	43.4	41.3
$\frac{11}{16}$	90.8	82.6	75.7	69.8	64.8	60.5	56.8	53.4	50.5	47.8	45.4
$\frac{3}{4}$	99.1	90.0	82.6	76.1	70.7	66.1	61.9	58.3	55.0	52.1	49.5
$\frac{13}{16}$	107.3	97.6	89.4	82.6	76.6	71.5	67.1	63.1	59.6	56.5	53.6
$\frac{7}{8}$	115.6	105.1	96.3	88.9	82.5	77.0	72.2	68.0	64.2	60.8	57.8
$\frac{15}{16}$	123.8	112.5	103.1	95.2	88.4	82.6	77.3	72.8	68.8	65.2	61.9
1	132.0	120.0	110.0	101.5	94.4	88.0	82.6	77.7	73.4	69.5	66.0
$1\frac{1}{8}$	148.6	135.1	123.8	114.3	106.1	99.1	92.8	87.4	82.5	78.2	74.3
$1\frac{1}{4}$	165.1	150.0	137.5	127.0	117.9	110.0	103.1	97.1	91.7	86.9	82.5
$1\frac{3}{8}$	181.6	165.1	151.3	139.7	129.7	121.0	113.5	106.9	100.9	95.6	90.8
$1\frac{1}{2}$	198.1	180.1	165.1	152.3	141.5	132.0	123.8	116.5	110.0	104.3	99.0

Table A-12.--Cylindrical Shells--External Pressure--Carbon Steel

Shell Thickness, Inches

(Computed from Fig. U-17 of the A.S.M.E. Code)

Outside Diameter, Inches	External Pressure, Lb. Per Sq. In.											
	15				30				50			
	Length of Shell, Ft.											
	2	4	6	8	2	4	6	8	2	4	6	8
24	.091	.125	.146	.164	.122	.163	.192	.216	.149	.199	.235	.264
30	.106	.141	.171	.183	.141	.184	.219	.246	.171	.225	.270	.300
36	.120	.155	.187	.216	.155	.209	.245	.257	.194	.252	.299	.338
42	.130	.168	.206	.285	.168	.222	.269	.298	.218	.273	.327	.370
48	.139	.182	.221	.250	.178	.245	.293	.327	.240	.298	.351	.399
54	.151	.200	.232	.270	.200	.265	.313	.357	.265	.324	.378	.432
60	.162	.213	.246	.282	.210	.282	.324	.372	.288	.342	.396	.450
66	.171	.224	.257	.297	.224	.297	.343	.396	.317	.363	.412	.475
72	.180	.238	.374	.310	.238	.310	.367	.418	.346	.389	.446	.504
78	.187	.250	.288	.324	.253	.327	.390	.429	.374	.413	.475	.530
84	.193	.260	.302	.336	.269	.336	.403	.445	.403	.436	.495	.545
90	.203	.270	.320	.351	.279	.342	.423	.468	.432	.459	.513	.567
96	.211	.278	.336	.365	.288	.365	.431	.490	.461	.480	.538	.595

Table A-12 (Cont'd.)--Cylindrical Shells--External Pressure--
Carbon Steel
Shell Thickness, Inches
(Computed from Fig. U-17 of the A.S.M.E. Code)

Outside Diameter, Inches	External Pressure, Lb. Per Sq. In.											
	75				100				125			
	Length of Shell, Ft.											
	2	4	6	8	2	4	6	8	2	4	6	8
24	.185	.235	.280	.312	.230	.259	.306	.346	.283	.297	.336	.381
30	.219	.267	.324	.360	.285	.306	.356	.396	.354	.360	.390	.426
36	.259	.299	.352	.396	.342	.356	.389	.438	.423	.423	.446	.485
42	.218	.332	.382	.436	.399	.407	.436	.486	.495	.495	.508	.537
48	.346	.370	.418	.465	.456	.460	.480	.518	.566	.566	.570	.595
54	.389	.405	.448	.508	.513	.513	.533	.566	.637	.637	.637	.663
60	.432	.438	.480	.534	.570	.570	.581	.611	.708	.708	.708	.720
66	.475	.475	.515	.561	.627	.627	.632	.660	.778	.778	.778	.778
72	.519	.519	.554	.597	.684	.684	.683	.711	.848	.848	.848	.848
78	.561	.561	.592	.631	.741	.741	.741	.764	.918	.918	.918	.918
84	.605	.605	.621	.663	.798	.798	.798	.813	.988	.988	.988	.988
90	.648	.648	.657	.702	.855	.855	.855	.862	1.060	1.060	1.060	1.060
96	.691	.691	.691	.740	.912	.912	.912	.920	1.130	1.130	1.130	1.130

Table A-13.--A.S.M.E. Flanged and Dished Heads

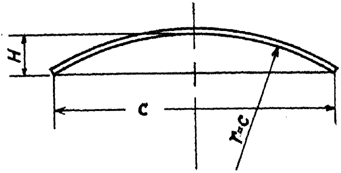
Approximate Surfaces and Volumes
Excluding straight flange

Diameter	Knuckle Radius	Radius of Dish	Surface	Volume	
Inches	Inches	Inches	Sq. Ft.	Cu. Ft.	Gal.
12	$\frac{3}{4}$	Same as Diameter	0 96	.09	.68
14	$\frac{7}{8}$		1.3	.14	1.06
16	1		1.8	.21	1.60
18	$1\frac{1}{8}$		2.2	.30	2.27
20	$1\frac{1}{4}$		2.7	.41	3.09
22	$1\frac{3}{8}$		3.2	.55	4.10
24	$1\frac{1}{2}$		3.8	.71	5.35
26	$1\frac{5}{8}$		4.5	.92	6.68
28	$1\frac{3}{4}$		5.2	1.14	8.52
30	$1\frac{7}{8}$		5.9	1.40	10.5
32	2		6.8	1.70	12.8
34	$2\frac{1}{8}$		7.6	2.04	15.2
36	$2\frac{1}{4}$		8.5	2.41	18.1
38	$2\frac{3}{8}$		9.5	2.84	21.3
40	$2\frac{1}{2}$		10.6	3.31	24.8
42	$2\frac{5}{8}$		11.7	3.84	28.8
48	$2\frac{7}{8}$		15.2	5.65	42.3
54	$3\frac{1}{4}$		19.5	8.0	60.1
60	$3\frac{5}{8}$		23.7	11.1	82.7
66	4		28.7	14.8	111.0
72	$4\frac{3}{8}$		34.2	19.1	144.0
78	$4\frac{3}{4}$		40.2	24.4	183.0
84	$5\frac{1}{8}$		46.7	30.5	229.0
90	$5\frac{1}{2}$		53.5	37.5	281.0
96	$5\frac{3}{4}$		60.9	45.5	341.0
102	$6\frac{1}{8}$		68.6	54.5	408.0
108	$6\frac{1}{2}$		77.0	64.6	484.0
114	$6\frac{7}{8}$		85.9	75.8	577.0
120	$7\frac{1}{4}$		95.3	88.4	662.0
126	$7\frac{5}{8}$		104.5	101.5	761.0
132	8		115.0	118.0	880.0

Table A-14--A.S.M.E. Elliptical Heads
Approximate Surfaces and Volumes
(Excluding Straight Flange)

Inside Diameter	Depth of Dish	Surface	Volume	
Inches	Inches	Sq. Ft.	Cu.In.	U.S. Gals.
24	6	4.35	1,809	7.84
30	7½	6.81	3,530	15.3
36	9	9.80	6,100	26.5
42	10½	13.32	9,700	42.1
48	12	17.42	14,500	62.9
54	13½	22.0	20,600	89.1
60	15	27.2	28,250	122.6
66	16½	32.9	37,600	163.0
72	18	39.2	48,800	212.0
78	19½	46.0	62,000	269.0
84	21	53.3	77,500	346.0
90	22½	61.2	95,500	414.0
96	24	69.6	115,800	502.0
102	25½	78.6	138,800	601.0
108	27	89.0	165,000	715.0
114	28½	98.1	194,000	841.0
120	30	108.8	226,000	979.0

Table A-15.- Dished-Only Heads
Approximate Surfaces and Volumes



Diameter C	Depth H	Surface	Volume	
Inches	Inches	Sq. Ft.	Cu. In.	U.S. Gals.
12	1.61	.84	92.9	.40
14	1.88	1.14	147.9	.64
16	2.15	1.49	220.5	.91
18	2.41	1.89	314	1.36
20	2.68	2.53	431	1.86
22	2.95	2.83	574	2.28
24	3.22	3.37	745	3.23
26	3.48	3.94	946	4.09
28	3.75	4.46	1,215	5.26
30	4.02	5.25	1,452	6.3
32	4.29	5.98	1,763	7.6
34	4.55	6.73	2,115	9.2
36	4.82	7.50	2,510	10.9
38	5.09	8.43	2,950	12.8
40	5.36	9.34	3,440	14.9
42	5.63	10.29	3,990	15.3
48	6.43	13.48	5,970	25.9
54	7.23	17.00	8,450	36.6
60	8.04	21.00	11,620	50.3
66	8.84	25.45	15,460	66.9
72	9.65	30.25	20,100	77.0
78	10.45	35.50	25,600	110.8
84	11.25	40.5	31,900	138.0
90	12.05	47.3	39,200	169.5
96	12.86	53.8	47,660	206.1
102	13.67	60.7	57,400	248
108	14.48	68.1	67,900	294
114	15.28	75.9	80,100	346
120	16.09	84.0	93,000	402
126	16.9	92.7	107,800	466
132	17.7	101.8	124,000	537

Table A-16. - Flanged and Dished Heads
A.S.M.E. Code--(A.P.I.--A.S.M.E.) Code Dimensions
(Bethlehem Steel Company)

All Dimensions are in Inches

O.D.	12			14			16			18			20			22			24			26		
Gage	Inside Knuckle Radii (K.R.), Inside Dish Radii (D.R.), and Approx. Max. Straight Flange (S.F.)																							
	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF
$\frac{3}{16}$	$\frac{3}{4}$	12	$1\frac{1}{2}$	$\frac{7}{8}$	14	$1\frac{1}{2}$	1	16	$1\frac{1}{2}$	$1\frac{1}{8}$	18	$1\frac{1}{2}$	$1\frac{1}{4}$	20	$1\frac{1}{2}$	$1\frac{3}{8}$	20	$1\frac{1}{2}$	$1\frac{1}{2}$	24	$1\frac{1}{2}$			
$\frac{1}{4}$	$\frac{3}{4}$	12	$2\frac{1}{2}$	$\frac{7}{8}$	14	$2\frac{1}{2}$	1	16	$2\frac{1}{2}$	$1\frac{1}{8}$	18	$2\frac{1}{2}$	$1\frac{1}{4}$	20	$2\frac{1}{2}$	$1\frac{3}{8}$	20	$2\frac{1}{2}$	$1\frac{1}{2}$	24	3	$1\frac{5}{8}$	24	3
$\frac{5}{16}$	$\frac{15}{16}$	12	3	$\frac{15}{16}$	14	3	1	16	3	$1\frac{1}{8}$	18	3	$1\frac{1}{4}$	20	3	$1\frac{3}{8}$	20	3	$1\frac{1}{2}$	24	$3\frac{1}{2}$	$1\frac{5}{8}$	24	$3\frac{1}{2}$
$\frac{3}{8}$	$1\frac{1}{8}$	12	3	$1\frac{1}{8}$	14	3	$1\frac{1}{8}$	16	3	$1\frac{1}{8}$	18	3	$1\frac{1}{4}$	20	$3\frac{1}{2}$	$1\frac{3}{8}$	20	$3\frac{1}{2}$	$1\frac{1}{2}$	24	4	$1\frac{5}{8}$	24	4
$\frac{7}{16}$	$1\frac{5}{16}$	12	$3\frac{1}{2}$	$1\frac{5}{16}$	14	$3\frac{1}{2}$	$1\frac{5}{16}$	16	$3\frac{1}{2}$	$1\frac{5}{16}$	18	$3\frac{1}{2}$	$1\frac{5}{16}$	17	4	$1\frac{3}{8}$	20	4	$1\frac{1}{2}$	24	5	$1\frac{5}{8}$	24	5
$\frac{1}{2}$	$1\frac{1}{2}$	10	$3\frac{1}{2}$	$1\frac{1}{2}$	12	$3\frac{1}{2}$	$1\frac{1}{2}$	16	$3\frac{1}{2}$	$1\frac{1}{2}$	15	$3\frac{1}{2}$	$1\frac{1}{2}$	17	4	$1\frac{1}{2}$	20	4	$1\frac{1}{2}$	24	5	$1\frac{5}{8}$	24	5
$\frac{5}{8}$	$1\frac{7}{8}$	10	$3\frac{1}{2}$	$1\frac{7}{8}$	12	4	$1\frac{7}{8}$	13	4	$1\frac{7}{8}$	15	4	$1\frac{7}{8}$	17	4	$1\frac{7}{8}$	19	4	$1\frac{7}{8}$	24	5	$1\frac{7}{8}$	24	5
$\frac{3}{4}$				$2\frac{1}{4}$	11	4	$2\frac{1}{4}$	13	4	$2\frac{1}{4}$	15	4	$2\frac{1}{4}$	17	4	$2\frac{1}{4}$	19	4	$2\frac{1}{4}$	21	5	$2\frac{1}{4}$	23	5
$\frac{7}{8}$							$2\frac{5}{8}$	12	4	$2\frac{5}{8}$	15	4	$2\frac{5}{8}$	15	4	$2\frac{5}{8}$	18	4	$2\frac{5}{8}$	21	5	$2\frac{5}{8}$	22	5
1										3	14	4	3	15	4	3	18	4	3	20	5	3	22	5
$1\frac{1}{8}$																			$3\frac{3}{8}$	19	5	$3\frac{3}{8}$	21	5

Minimum S.F. Dimension, $1\frac{1}{2}$" in all cases. S.F. may be any dimension between maximum and minimum.
Heads can be furnished with larger or special K.R. dimensions and heavier gauges.
All dimensions subject to usual shop tolerances (within A.S.M.E. Code requirements).

Table A-16 (Cont'd). - Flanged and Dished Heads
A.S.M.E. Code--(A.P.I.--A.S.M.E.) Code Dimensions
(Bethlehem Steel Company)

All Dimensions are in Inches

O.D.	28			30			32			34			36			38			40			42		
Gage	Inside Knuckle Radii (K.R.), Inside Dish Radii (D.R.), and Approx. Max. Straight Flange (S.F.)																							
	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF
1/4	1 3/4	24	3	1 7/8	30	3	2	30	3	2 1/8	30	3	2 1/4	36	3									
5/16	1 3/4	24	3 1/2	1 7/8	30	3 1/2	2	30	3 1/2	2 1/8	30	3 1/2	2 1/4	36	3 1/2	2 3/8	36	3 1/2	2 1/2	36	3 1/2	2 5/8	42	3 1/2
3/8	1 3/4	24	4	1 7/8	30	4	2	30	4	2 1/8	30	4	2 1/4	36	4 1/2	2 3/8	36	4 1/2	2 1/2	36	4 1/2	2 5/8	42	4 1/2
7/16	1 3/4	24	5	1 7/8	30	5	2	30	5	2 1/8	30	5	2 1/4	36	6	2 3/8	36	6	2 1/2	36	6	2 5/8	42	6
1/2	1 3/4	24	5	1 7/8	30	5	2	30	5	2 1/8	30	5	2 1/4	36	6	2 3/8	36	6	2 1/2	36	6	2 5/8	42	6
5/8	1 7/8	24	5	1 7/8	30	5	2	30	5	2 1/8	30	5	2 1/4	36	6	2 3/8	36	6	2 1/2	36	6	2 5/8	42	6
3/4	2 1/4	24	5	2 1/4	24	5	2 1/4	30	5	2 1/4	30	5	2 1/4	36	6	2 3/8	36	6	2 1/2	36	6	2 5/8	42	6
7/8	2 5/8	24	5	2 5/8	24	5	2 5/8	30	5	2 5/8	30	5	2 5/8	30	6	2 5/8	36	6	2 5/8	36	6	2 5/8	42	6
1	3	24	5	3	24	5	3	30	5	3	30	5	3	30	6	3	36	6	3	36	6	3	42	6
1 1/8	3 3/8	23	5	3 3/8	24	5	3 3/8	30	5	3 3/8	30	5	3 3/8	30	6	3 3/8	36	6	3 3/8	36	6	3 3/8	36	6
1 1/4				3 3/4	24	5	3 3/4	30	5	3 3/4	30	5	3 3/4	30	6	3 3/4	36	6	3 3/4	36	6	3 3/4	36	6
1 3/8				4 1/8	24	5	4 1/8	24	5	4 1/8	30	5	4 1/8	30	6	4 1/8	30	6	4 1/8	36	6	4 1/8	36	6
1 1/2				4 1/2	23	5	4 1/2	24	5	4 1/2	30	5	4 1/2	30	6	4 1/2	30	6	4 1/2	36	6	4 1/2	36	6
1 5/8													4 7/8	30	6	4 7/8	30	6	4 7/8	30	6	4 7/8	36	6
1 3/4													5 1/4	30	6	5 1/4	30	6	5 1/4	30	6	5 1/4	36	6
1 7/8													5 5/8	24	6	5 5/8	30	6	5 5/8	30	6	5 5/8	36	6
2																						6	30	6
2 1/4																						6 3/4	30	6

Table A-16 (Cont'd). - Flanged and Dished Heads
A.S.M.E. Code--(A.P.I.--A.S.M.E.) Code Dimensions
(Bethlehem Steel Company)
All Dimensions are in Inches

O.D.	48			54			60			66			72			78			84			90		
Gage	Inside Knuckle Radii (K.R.), Inside Dish Radii (D.R.), and Approx. Max. Straight Flange (S.F.)																							
	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF
$\frac{5}{16}$	$2\frac{7}{8}$	48	$3\frac{1}{2}$	$3\frac{1}{4}$	54	$3\frac{1}{2}$																		
$\frac{3}{8}$	$2\frac{7}{8}$	48	$4\frac{1}{2}$	$3\frac{1}{4}$	48	$4\frac{1}{2}$	$3\frac{5}{8}$	54	$4\frac{1}{2}$	4	60	$4\frac{1}{2}$	$4\frac{3}{8}$	66	$4\frac{1}{2}$	$4\frac{3}{4}$	72	$4\frac{1}{2}$						
$\frac{7}{16}$	$2\frac{7}{8}$	48	6	$3\frac{1}{4}$	48	6	$3\frac{5}{8}$	54	6	4	60	6	$4\frac{3}{8}$	66	6	$4\frac{3}{4}$	72	6	$5\frac{1}{8}$	78	6	$5\frac{1}{2}$	84	6
$\frac{1}{2}$	$2\frac{7}{8}$	48	6	$3\frac{1}{4}$	48	6	$3\frac{5}{8}$	54	6	4	60	6	$4\frac{3}{8}$	66	6	$4\frac{3}{4}$	72	6	$5\frac{1}{8}$	78	6	$5\frac{1}{2}$	84	6
$\frac{5}{8}$	$2\frac{7}{8}$	48	6	$3\frac{1}{4}$	48	6	$3\frac{5}{8}$	54	6	4	60	6	$4\frac{3}{8}$	66	6	$4\frac{3}{4}$	72	6	$5\frac{1}{8}$	78	6	$5\frac{1}{2}$	84	6
$\frac{3}{4}$	$2\frac{7}{8}$	48	6	$3\frac{1}{4}$	48	6	$3\frac{5}{8}$	54	6	4	60	6	$4\frac{3}{8}$	66	7	$4\frac{3}{4}$	72	7	$5\frac{1}{8}$	78	7	$5\frac{1}{2}$	84	7
$\frac{7}{8}$	$2\frac{7}{8}$	48	6	$3\frac{1}{4}$	48	6	$3\frac{5}{8}$	54	6	4	60	6	$4\frac{3}{8}$	66	7	$4\frac{3}{4}$	72	7	$5\frac{1}{8}$	78	7	$5\frac{1}{2}$	84	7
1	3	42	7	$3\frac{1}{4}$	48	7	$3\frac{5}{8}$	54	7	4	60	7	$4\frac{3}{8}$	66	8	$4\frac{3}{4}$	72	8	$5\frac{1}{8}$	78	8	$5\frac{1}{2}$	84	8
$1\frac{1}{8}$	$3\frac{3}{8}$	42	7	$3\frac{3}{8}$	48	7	$3\frac{5}{8}$	54	7	4	60	7	$4\frac{3}{8}$	66	8	$4\frac{3}{4}$	72	8	$5\frac{1}{8}$	78	8	$5\frac{1}{2}$	84	8
$1\frac{1}{4}$	$3\frac{3}{4}$	42	8	$3\frac{3}{4}$	48	8	$3\frac{3}{4}$	54	8	4	60	8	$4\frac{3}{8}$	66	8	$4\frac{3}{4}$	72	8	$5\frac{1}{8}$	78	8	$5\frac{1}{2}$	84	8
$1\frac{3}{8}$	$4\frac{1}{8}$	42	8	$4\frac{1}{8}$	48	8	$4\frac{1}{8}$	54	8	$4\frac{1}{8}$	60	8	$4\frac{3}{8}$	66	8	$4\frac{3}{4}$	72	8	$5\frac{1}{8}$	78	8	$5\frac{1}{2}$	84	8
$1\frac{1}{2}$	$4\frac{1}{2}$	42	8	$4\frac{1}{2}$	48	8.	$4\frac{1}{2}$	54	8	$4\frac{1}{2}$	60	8	$4\frac{1}{2}$	66	8	$4\frac{3}{4}$	72	8	$5\frac{1}{8}$	78	8	$5\frac{1}{2}$	84	8
$1\frac{5}{8}$	$4\frac{7}{8}$	42	8	$4\frac{7}{8}$	48	8	$4\frac{7}{8}$	54	8	$4\frac{7}{8}$	60	8	$4\frac{7}{8}$	66	8	$4\frac{7}{8}$	72	8	$5\frac{1}{8}$	78	8	$5\frac{1}{2}$	84	8
$1\frac{3}{4}$	$5\frac{1}{4}$	42	8	$5\frac{1}{4}$	48	8	$5\frac{1}{4}$	54	8	$5\frac{1}{4}$	60	8	$5\frac{1}{4}$	66	8	$5\frac{1}{4}$	72	8	$5\frac{1}{4}$	78	8	$5\frac{1}{2}$	84	8
$1\frac{7}{8}$	$5\frac{5}{8}$	42	8	$5\frac{5}{8}$	42	8	$5\frac{5}{8}$	48	8	$5\frac{5}{8}$	54	8	$5\frac{5}{8}$	60	8	$5\frac{5}{8}$	66	8	$5\frac{5}{8}$	72	8	$5\frac{5}{8}$	78	8
2	6	36	8	6	42	8	6	48	8	6	54	8	6	60	8	6	66	8	6	72	8	6	78	8
$2\frac{1}{4}$	$6\frac{3}{4}$	36	8	$6\frac{3}{4}$	42	8	$6\frac{3}{4}$	48	8	$6\frac{3}{4}$	54	8	$6\frac{3}{4}$	60	8	$6\frac{3}{4}$	66	8	$6\frac{3}{4}$	72	8	$6\frac{3}{4}$	78	8
$2\frac{1}{2}$	$7\frac{1}{2}$	36	8	$7\frac{1}{2}$	42	8	$7\frac{1}{2}$	48	8	$7\frac{1}{2}$	54	8	$7\frac{1}{2}$	60	8	$7\frac{1}{2}$	66	8	$7\frac{1}{2}$	72	8	$7\frac{1}{2}$	78	8

Table A-16 (Cont'd). - Flanged and Dished Heads
A.S.M.E. Code--(A.P.I.--A.S.M.E.) Code Dimensions
(Bethlehem Steel Company)

All Dimensions are in Inches

O.D.	96			102			108			114			120			126			132					
Gage	Inside Knuckle Radii (K.R.), Inside Dish Radii (D.R.), and Approx. Max. Straight Flange (S.F.)																							
	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF	KR	DR	SF
7/16	5 3/4	90	6																					
1/2	5 3/4	90	6	6 1/8	96	6	6 1/2	102	6	6 7/8	108	6	7 1/4	114	6									
5/8	5 3/4	90	6	6 1/8	96	6	6 1/2	102	6	6 7/8	108	6	7 1/4	114	6									
3/4	5 3/4	90	7	6 1/8	96	7	6 1/2	102	7	6 7/8	108	7	7 1/4	114	7	7 5/8	120	7	8	126	4			
7/8	5 3/4	90	7	6 1/8	96	7	6 1/2	102	7	6 7/8	108	7	7 1/4	114	7	7 5/8	120	7	8	126	4			
1	5 3/4	90	8	6 1/8	96	8	6 1/2	102	8	6 7/8	108	8	7 1/4	114	8	7 5/8	120	7 1/2	8	126	4			
1 1/8	5 3/4	90	8	6 1/8	96	8	6 1/2	102	8	6 7/8	108	8	7 1/4	114	8	7 5/8	120	7 1/2	8	120	4			
1 1/4	5 3/4	90	8	6 1/8	96	8	6 1/2	102	8	6 7/8	108	8	7 1/4	114	8	7 5/8	120	7 1/2	8	120	4			
1 3/8													7 1/4	108	8	7 5/8	114	7 1/2	8	120	4			
1 1/2	5 3/4	90	8	6 1/8	96	8	6 1/2	102	8	6 7/8	108	8	7 1/4	108	8	7 5/8	114	7 1/2	8	120	4			
1 5/8	5 3/4	90	8	6 1/8	96	8	6 1/2	96	8	6 7/8	102	8	7 1/4	102	8	7 5/8	114	7 1/2	8	120	4			
1 3/4	5 3/4	90	8	6 1/8	96	8	6 1/2	[illegible]	8	6 7/8	102	8	7 1/4	102	8	7 5/8	114	7 1/2	8	120	4			
1 7/8	5 3/4	[illegible]	8	6 1/8	[illegible]	8	6 1/2	[illegible]	8	6 7/8	102	8	7 1/4	102	8	7 5/8	114	7 1/2	8	120	4			
2	[illegible]	[illegible]	8	6 1/8	[illegible]	8	6 1/2	[illegible]	8	[illegible]	102	8	7 1/4	102	8	7 5/8	114	7 1/2	8	120	4			
2 1/4	5 3/4	[illegible]	8	6 1/8	[illegible]	8	6 1/2	[illegible]	8	[illegible]	102	8	7 1/4	102	8	7 5/8	114	7 1/2	8	120	4			
2 1/2	[illegible]	[illegible]	8	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	102	[illegible]	[illegible]	102	8	7 5/8	114	7 1/2	8	120	4			

[illegible]

Table A-17. - Elliptical Heads
A.S.M.E. Code and A.P.I.-A.S.M.E. Code
Dimensions

(Bethlehem Steel Company)

Dimensions in Inches

I.D.	Gage T		Maximum S.F. for		Depth = ¼ of I.D.
	Minimum	Maximum	Minimum T	Maximum T	
24	5/16	1½	4½	4	6
30	3/8	2	3½	7	7½
36	3/8	2½	3½	7	9
42	3/8	3	3½	10	10½
48	3/8	3	3½	10	12
54	½	3	4	10	13½
60	½	3	4	10	15
66	½	3	4	10	16½
72	½	3	4	10	18
78	9/16	3	5	10	19½
84	9/16	3	5	10	21
90	9/16	3	5	10	22½
96	9/16	3	5	8	24
102	9/16	3	5	8	25½
108	9/16	3	5	8	27
114	5/8	2	5	5	28½
120	5/8	2	2	2	30

All dimensions subject to usual shop tolerances (within A.S.M.E. and A.P.I.--A.S.M.E. Code requirements).
Maximum S.F. dimension for intermediate gages is proportional.

Table A-18.--U. S. Gallons in Round Tanks

For One Foot in Depth

Dia. of Tanks ft.	in.	No. U. S. Gals.	Cu. Ft. and Area in Sq. Ft.
1		5.87	.785
1	1	6.89	.922
1	2	8.00	1.069
1	3	9.18	1.227
1	4	10.44	1.396
1	5	11.79	1.576
1	6	13.22	1.767
1	7	14.73	1.969
1	8	16.32	2.182
1	9	17.99	2.405
1	10	19.75	2.640
1	11	21.58	2.885
2		23.50	3.142
2	1	25.50	3.409
2	2	27.58	3.687
2	3	29.74	3.976
2	4	31.99	4.276
2	5	34.31	4.587
2	6	36.72	4.909
2	7	39.21	5.241
2	8	41.78	5.585
2	9	44.43	5.940
2	10	47.16	6.305
2	11	49.98	6.681
3		52.88	7.069
3	1	55.86	7.467
3	2	58.92	7.876
3	3	62.06	8.296
3	4	65.28	8.727
3	5	68.58	9.168
3	6	71.97	9.621
3	7	75.44	10.085
3	8	78.99	10.559
3	9	82.62	11.045
3	10	86.33	11.541
3	11	90.13	12.048
4		94.	12.566
4	1	97.96	13.095
4	2	102.	13.635
4	3	106.12	14.186
4	4	110.32	14.748
4	5	114.61	15.321
4	6	118.97	15.90
4	7	123.42	16.50
4	8	127.95	17.10
4	9	132.56	17.72
4	10	137.25	18.35
4	11	142.02	18.99
5		146.88	19.63
5	1	151.82	20.29
5	2	156.83	20.97
5	3	161.93	21.65
5	4	167.12	22.34
5	5	172.38	23.04
5	6	177.72	23.76
5	7	183.15	24.48
5	8	188.66	25.22
5	9	194.25	25.97
5	10	199.92	26.73
5	11	205.67	27.49
6		211.51	28.27
6	3	229.50	30.68
6	6	248.23	33.18
6	9	267.69	35.78
7		287.88	38.48
7	3	308.81	41.28
7	6	330.48	44.18
7	9	352.88	47.17
8		376.01	50.27
8	3	399.88	53.46
8	6	424.48	56.75
8	9	449.82	60.13
9		475.89	63.62
9	3	502.70	67.20
9	6	530.24	70.88
9	9	558.51	74.66
10		587.52	78.54
10	3	617.26	82.52
10	6	640.74	86.59
10	9	678.95	90.76
11		710.90	95.03
11	3	743.58	99.40
11	6	776.99	103.87
11	9	811.14	108.43
12		846.03	113.10
12	3	881.65	117.86
12	6	918.	122.72
12	9	955.09	127.68
13		992.91	132.73
13	3	1031.50	137.89
13	6	1070.80	143.14
13	9	1110.80	148.49
14		1151.50	153.94
14	3	1193.	159.48
14	6	1235.30	165.13
14	9	1278.20	170.87
15		1321.90	176.71
15	3	1366.40	182.65
15	6	1411.50	188.69
15	9	1457.40	194.83
16		1504.10	201.06
16	3	1551.40	207.39
16	6	1599.50	213.82
16	9	1648.40	220.35
17		1697.90	226.98
17	3	1748.20	233.71
17	6	1799.30	240.53
17	9	1851.10	247.45
18		1903.60	254.47
18	3	1956.80	261.59
18	6	2010.80	268.80
18	9	2065.50	276.12
19		2120.90	283.53
19	3	2177.10	291.04
19	6	2234.	298.65
19	9	2291.70	306.35
20		2350.10	314.16
20	3	2409.20	322.06
20	6	2469.10	330.06
20	9	2529.60	338.16
21		2591.	346.36
21	3	2653.	354.66
21	6	2715.80	363.05
21	9	2779.30	371.54
22		2843.60	380.13
22	3	2908.60	388.82
22	6	2974.30	397.61
22	9	3040.80	406.49
23		3108.	415.48
23	3	3175.90	424.56
23	6	3244.60	433.74
23	9	3314.	443.01
24		3384.10	452.39
24	3	3455.	461.86
24	6	3526.60	471.44
24	9	3598.90	481.11
25		3672.	490.87
25	3	3745.80	500.74
25	6	3820.30	510.71
25	9	3895.60	520.77
26		3971.60	530.93
26	3	4048.40	541.19
26	6	4125.90	551.55
26	9	4204.10	562.
27		4283.	572.66
27	3	4362.70	583.21
27	6	4443.10	593.96
27	9	4524.30	604.81
28		4606.20	615.75
28	3	4688.80	626.80
28	6	4772.10	637.94
28	9	4856.20	649.18
29		4941.	660.52
29	3	5026.60	671.96
29	6	5112.90	683.49
29	9	5199.90	695.13
30		5287.70	706.86
30	3	5376.20	718.69
30	6	5465.40	730.62
30	9	5555.40	742.64
31		5646.10	754.77
31	3	5737.50	766.99
31	6	5829.70	779.31
31	9	5922.60	791.73
32		6016.20	804.25
32	3	6110.60	816.86
32	6	6205.70	829.58
32	9	6301.50	842.39

To find the capacity of tanks greater than the largest given in the table, look in the table for a tank of one-half of the given size and multiply its capacity by 4, or one of one-third its size and multiply its capacity by 9, etc.

Table A-19.--Number of U. S. Gallons in Rectangular Tanks

For one foot in depth

Width of Tank	Length of Tank ft. 2	ft. in. 2 6	ft. 3	ft. in. 3 6	ft. 4	ft. in. 4 6	ft. 5	ft. in. 5 6	ft. 6	ft. in. 6 6	ft. 7
2 ft.	29.92	37.40	44.88	52.36	59.84	67.32	74.81	82.29	89.77	97.25	104.73
2 6"		46.75	56.10	65.45	74.80	84.16	93.51	102.86	112.21	121.56	130.91
3			67.32	78.54	89.77	100.99	112.21	123.43	134.65	145.87	157.09
3 6"				91.64	104.73	117.82	130.91	144.00	157.09	170.18	183.27
4					119.69	134.65	149.61	164.57	179.53	194.49	209.45
4 6"						151.48	168.31	185.14	201.97	218.80	235.63
5							187.01	205.71	224.41	243.11	261.82
5 6"								226.28	246.86	267.43	288.00
6									269.30	291.74	314.18
6 6"										316.05	340.36
7											366.54

Table A-19.--Number of U. S. Gallons in Rectangular Tanks (Cont'd.)

For one foot in depth

Width of Tank	Length of Tank ft. in. 7 6	ft. 8	ft. in. 8 6	ft. 9	ft. in. 9 6	ft. 10	ft. in. 10 6	ft. 11	ft. in. 11 6	ft. 12
2 ft.	112.21	119.69	127.17	134.65	142.13	149.61	157.09	164.57	172.05	179.53
2 6"	140.26	149.61	158.96	168.31	177.66	187.01	196.36	205.71	215.06	224.41
3	168.31	179.53	190.75	202.97	213.19	224.41	235.63	246.86	258.07	269.30
3 6"	196.36	209.45	222.54	235.63	248.73	261.82	274.90	288.00	301.09	314.18
4	224.41	239.37	254.34	269.30	284.26	299.22	314.18	329.14	344.10	359.06
4 6"	252.47	269.30	286.13	302.96	319.79	336.62	353.45	370.28	387.11	403.94
5	280.52	299.22	317.92	336.62	355.32	374.03	392.72	411.43	430.13	448.83
5 6"	308.57	329.14	349.71	370.28	390.85	411.43	432.00	452.57	473.14	493.71
6	336.62	359.06	381.50	403.94	426.39	448.83	471.27	493.71	516.15	538.59
6 6"	364.67	388.98	413.30	437.60	461.92	486.23	510.54	534.85	559.16	583.47
7	392.72	418.91	445.09	471.27	497.45	523.64	549.81	575.99	602.18	628.36
7 6"	420.78	448.83	476.88	504.93	532.98	561.04	589.08	617.14	645.19	673.24
8		478.75	508.67	538.59	568.51	598.44	628.36	658.28	688.20	718.12
8 6"			540.46	572.25	604.05	635.84	667.63	699.42	731.21	763.00
9				605.92	639.58	673.25	706.90	740.56	774.23	807.89
9 6"					675.11	710.65	746.17	781.71	817.24	852.77
10						748.05	785.45	822.86	860.26	897.66
10 6"							824.73	864.00	903.26	942.56
11								905.14	946.27	987.43
11 6"									989.29	1032.3
12										1077.2

Table A-20.--Standard Wire Gauges

(1) Birmingham or Stub's,
(2) American or Brown and Sharp,
(3) U. S. Standard,
(4) Washburn and Moen,
(5) Music Wire (Std.).

Aluminum (2); except tubing (1)
Bands (1)
Brass Tubing: (3/8" O.D. and larger) (1)
(Smaller than 3/8" O.D.) (2)
Brass Sheets (2)
Brass Strips (2)
Copper Sheets (2)
Copper Wire (2)
Flat Wire (1)
Hoops (1)
Iron Wire (4)
Stainless Steel Sheets (3)
Music Wire (5)
Nickel Sheets (2)
Nickel Silver Wire (2)
Phosphor Bronze Strip (2)
Spring Steel (1)
Steel Plates (3)
Steel Sheets (3)
Steel Tubing, seamless and welded, (1)
Steel Wire (4); exceptions: Music Wire (5), Armature Binding Wire (2), Flat Wire (5)
Strip Steel (1)

Table A-21.--Dimensions of Welded and Seamless Steel Pipe

(Tentative A.S.A. Standard B36.10)

Nominal Pipe Size	Outside Diam.	Nominal Wall Thicknesses for Schedule Numbers									
		Sched. 10	Sched. 20	Sched. 30	Sched. 40	Sched. 60	Sched. 80	Sched. 100	Sched. 120	Sched. 140	Sched. 160
⅛	0.405	......	......	......	**0.068**	......	**0.095**	......	......	......	......
¼	0.540	......	......	......	**0.088**	......	**0.119**	......	......	......	......
⅜	0.675	......	......	......	**0.091**	......	**0.126**	......	......	......	......
½	0.840	......	......	......	**0.109**	......	**0.147**	......	......	......	0.187
¾	1.050	......	......	......	**0.113**	......	**0.154**	......	......	......	0.218
1	1.315	......	......	......	**0.133**	......	**0.179**	......	......	......	0.250
1¼	1.660	......	......	......	**0.140**	......	**0.191**	......	......	......	0.250
1½	1.900	......	......	......	**0.145**	......	**0.200**	......	......	......	0.281
2	2.375	......	......	......	**0.154**	......	**0.218**	......	......	......	0.343
2½	2.875	......	......	......	**0.203**	......	**0.276**	......	......	......	0.375
3	3.5	......	......	......	**0.216**	......	**0.300**	......	......	......	0.437
3½	4.0	......	......	......	**0.226**	......	**0.318**	......	......	......	......
4	4.5	......	......	......	**0.237**	......	**0.337**	......	0.437	......	0.531
5	5.563	......	......	......	**0.258**	......	**0.375**	......	0.500	......	0.625
6	6.625	......	......	......	**0.280**	......	**0.432**	......	0.562	......	0.718
8	8.625	......	0.250	**0.277**	**0.322**	0.406	**0.500**	0.593	0.718	0.812	0.906
10	10.75	......	0.250	**0.307**	**0.365**	**0.500**	0.593	0.718	0.843	1.000	1.125
12	12.75	......	0.250	**0.330**	0.406	0.562	0.687	0.843	1.000	1.125	1.312
14 O.D.	14.0	0.250	0.312	0.375	0.437	0.593	0.750	0.937	1.062	1.250	1.406
16 O.D.	16.0	0.250	0.312	0.375	0.500	0.656	0.843	1.031	1.218	1.437	1.562
18 O.D.	18.0	0.250	0.312	0.437	0.562	0.718	0.937	1.156	1.343	1.562	1.750
20 O.D.	20.0	0.250	0.375	0.500	0.593	0.812	1.031	1.250	1.500	1.750	1.937
24 O.D.	24.0	0.250	0.375	0.562	0.687	0.937	1.218	1.500	1.750	2.062	2.312
30 O.D.	30.0	0.312	0.500	0.625	......	......	......	......	......	......	......

All dimensions are given in inches.

Thicknesses shown in bold face type in Schedules 30 and 40 are identical with thicknesses for "standard weight" pipe in former lists; those in Schedules 60 and 80 are identical with thicknesses for "extra strong" pipe in former lists.

The Schedule Numbers indicate approximate values of the expression 1000 × P/S.

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions. For tolerances on wall thicknesses, see appropriate material specification.

Table A-22. - Length of Thread on Pipe That is screwed into valves or fittings to make a tight joint

Size Inches	Size Millimeters	Length of Engagement Inches	Size Inches	Size Millimeters	Length of Engagement Inches
$\frac{1}{8}$	3	$\frac{5}{16}$	$3\frac{1}{2}$	90	$1\frac{3}{16}$
$\frac{1}{4}$	6	$\frac{7}{16}$	4	100	$1\frac{3}{16}$
$\frac{3}{8}$	10	$\frac{7}{16}$	$4\frac{1}{2}$	113	$1\frac{1}{4}$
$\frac{1}{2}$	13	$\frac{9}{16}$	5	125	$1\frac{5}{16}$
$\frac{3}{4}$	19	$\frac{9}{16}$	6	150	$1\frac{3}{8}$
1	25	$\frac{11}{16}$	7	175	$1\frac{3}{8}$
$1\frac{1}{4}$	32	$\frac{11}{16}$	8	200	$1\frac{7}{16}$
$1\frac{1}{2}$	38	$\frac{11}{16}$	9	225	$1\frac{1}{2}$
2	50	$\frac{3}{4}$	10	250	$1\frac{5}{8}$
$2\frac{1}{2}$	64	$1\frac{1}{16}$	12	300	$1\frac{3}{4}$
3	76	$1\frac{1}{8}$			

Table A-23. - Templates for Drilling

150-Pound Bronze Flanges

Pipe Size Inches	Flange Diameter Inches	Flange Thickness Inches	Bolt Circle Inches	Number of Bolts	Diameter and Length of Bolts Inches	Length of Bolt Studs with 2 Nuts Inches
$\frac{1}{2}$	$3\frac{1}{2}$	$\frac{5}{16}$	$2\frac{3}{8}$	4	$\frac{1}{2}$ x $1\frac{1}{4}$	$1\frac{7}{8}$
$\frac{3}{4}$	$3\frac{7}{8}$	$\frac{11}{32}$	$2\frac{3}{4}$	4	$\frac{1}{2}$ x $1\frac{1}{2}$	$1\frac{7}{8}$
1	$4\frac{1}{4}$	$\frac{3}{8}$	$3\frac{1}{8}$	4	$\frac{1}{2}$ x $1\frac{1}{2}$	2
$1\frac{1}{4}$	$4\frac{5}{8}$	$\frac{13}{32}$	$3\frac{1}{2}$	4	$\frac{1}{2}$ x $1\frac{1}{2}$	2
$1\frac{1}{2}$	5	$\frac{7}{16}$	$3\frac{7}{8}$	4	$\frac{1}{2}$ x $1\frac{1}{2}$	$2\frac{1}{8}$
2	6	$\frac{1}{2}$	$4\frac{3}{4}$	4	$\frac{5}{8}$ x $1\frac{3}{4}$	$2\frac{1}{2}$
$2\frac{1}{2}$	7	$\frac{9}{16}$	$5\frac{1}{2}$	4	$\frac{5}{8}$ x 2	$2\frac{5}{8}$
3	$7\frac{1}{2}$	$\frac{5}{8}$	6	4	$\frac{5}{8}$ x 2	$2\frac{3}{4}$
$3\frac{1}{2}$	$8\frac{1}{2}$	$\frac{11}{16}$	7	8	$\frac{5}{8}$ x $2\frac{1}{4}$	$2\frac{7}{8}$
4	9	$\frac{11}{16}$	$7\frac{1}{2}$	8	$\frac{5}{8}$ x $2\frac{1}{4}$	$2\frac{7}{8}$
5	10	$\frac{3}{4}$	$8\frac{1}{2}$	8	$\frac{3}{4}$ x $2\frac{1}{2}$	$3\frac{1}{4}$
6	11	$\frac{13}{16}$	$9\frac{1}{2}$	8	$\frac{3}{4}$ x $2\frac{1}{2}$	$3\frac{3}{8}$
8	$13\frac{1}{2}$	$\frac{15}{16}$	$11\frac{3}{4}$	8	$\frac{3}{4}$ x $2\frac{3}{4}$	$3\frac{5}{8}$
10	16	1	$14\frac{1}{4}$	12	$\frac{7}{8}$ x $3\frac{1}{4}$	$4\frac{1}{8}$
12	19	$1\frac{1}{16}$	17	12	$\frac{7}{8}$ x $3\frac{1}{4}$	$4\frac{1}{4}$

Bolt holes are drilled $\frac{1}{8}$ inch larger than the bolt diameters.

Table A-24. - Templates for Drilling

300-Pound Bronze Flanges

Pipe Size Inches	Flange Diameter Inches	Flange Thickness Inches	Bolt Circle Inches	Number of Bolts	Diameter and Length of Bolts Inches	Length of Bolt Studs with 2 Nuts Inches
$\frac{1}{2}$	$3\frac{3}{4}$	$\frac{1}{2}$	$2\frac{5}{8}$	4	$\frac{1}{2} \times 1\frac{3}{4}$	$2\frac{1}{4}$
$\frac{3}{4}$	$4\frac{5}{8}$	$\frac{17}{32}$	$3\frac{1}{4}$	4	$\frac{5}{8} \times 2$	$2\frac{1}{2}$
1	$4\frac{7}{8}$	$\frac{19}{32}$	$3\frac{1}{2}$	4	$\frac{5}{8} \times 2$	$2\frac{5}{8}$
$1\frac{1}{4}$	$5\frac{1}{4}$	$\frac{5}{8}$	$3\frac{7}{8}$	4	$\frac{5}{8} \times 2$	$2\frac{3}{4}$
$1\frac{1}{2}$	$6\frac{1}{8}$	$\frac{11}{16}$	$4\frac{1}{2}$	4	$\frac{3}{4} \times 2\frac{1}{4}$	$3\frac{1}{8}$
2	$6\frac{1}{2}$	$\frac{3}{4}$	5	8	$\frac{5}{8} \times 2\frac{1}{4}$	3
$2\frac{1}{2}$	$7\frac{1}{2}$	$\frac{13}{16}$	$5\frac{7}{8}$	8	$\frac{3}{4} \times 2\frac{1}{2}$	$3\frac{3}{8}$
3	$8\frac{1}{4}$	$\frac{29}{32}$	$6\frac{5}{8}$	8	$\frac{3}{4} \times 2\frac{3}{4}$	$3\frac{1}{2}$
$3\frac{1}{2}$	9	$\frac{31}{32}$	$7\frac{1}{4}$	8	$\frac{3}{4} \times 3$	$3\frac{5}{8}$
4	10	$1\frac{1}{16}$	$7\frac{7}{8}$	8	$\frac{3}{4} \times 3$	$3\frac{7}{8}$
5	11	$1\frac{1}{8}$	$9\frac{1}{4}$	8	$\frac{3}{4} \times 3\frac{1}{4}$	4
6	$12\frac{1}{2}$	$1\frac{3}{16}$	$10\frac{5}{8}$	12	$\frac{3}{4} \times 3\frac{1}{4}$	$4\frac{1}{8}$
8	15	$1\frac{3}{8}$	13	12	$\frac{7}{8} \times 3\frac{3}{4}$	$4\frac{3}{4}$
10	$17\frac{1}{2}$	$1\frac{7}{16}$	...	...	...	...
12	$20\frac{1}{2}$	$1\frac{17}{32}$	...	...	...	...

Bolt holes are drilled $\frac{1}{8}$ inch larger than the bolt diameters.

Table A-25. - Templates for Drilling

25-Pound (Low Pressure) Cast Iron Flanges and Fittings
A.S.A. B 16 b2 - 1931

Pipe Size Inches	Flange Diameter Inches	Flange Thickness Inches	Bolt Circle Inches	Number of Bolts	Diameter and Length of Bolts Inches
4	9	¾	7½	8	⅝ x 2¼
5	10	¾	8½	8	⅝ x 2¼
6	11	¾	9½	8	⅝ x 2¼
8	13½	¾	11¾	8	⅝ x 2¼
10	16	⅞	14¼	12	⅝ x 2½
12	19	1	17	12	⅝ x 2¾
14	21	1⅛	18¾	12	¾ x 3¼
16	23½	1⅛	21¼	16	¾ x 3¼
18	25	1¼	22¾	16	¾ x 3½
20	27½	1¼	25	20	¾ x 3½
24	32	1⅜	29½	20	¾ x 3¾
30	38¾	1½	36	28	⅞ x 4¼
36	46	1⅝	42¾	32	⅞ x 5
42	53	1¾	49½	36	1 x 5¼
48	59½	2	56	44	1 x 5½
54	66¼	2¼	62¾	44	1 x 5¾
60	73	2¼	69¼	52	1⅛ x 6
72	86½	2½	82½	60	1⅛ x 6¼
84	99¾	2¾	95½	64	1¼ x 7¼
96	113¼	3	108½	68	1¼ x 7¾

All 25-pound cast iron standard flanges have plain faces.
Bolt holes are drilled ⅛ inch larger than diameter of bolts.

Table A-26. - Templates for Drilling

125-Pound (Standard) Cast Iron Flanges and Fittings
A.S.A. B 16a - 1931

Size Inches	Flange Diameter Inches	Flange Thickness Inches	Bolt Circle Inches	Number of Bolts	Diameter and Length of Bolts Inches	Length of Bolt Studs with 2 Nuts Inches
1	$4\frac{1}{4}$	$\frac{7}{16}$	$3\frac{1}{8}$	4	$\frac{1}{2} \times 1\frac{1}{2}$	...
$1\frac{1}{4}$	$4\frac{5}{8}$	$\frac{1}{2}$	$3\frac{1}{2}$	4	$\frac{1}{2} \times 1\frac{1}{2}$	...
$1\frac{1}{2}$	5	$\frac{9}{16}$	$3\frac{7}{8}$	4	$\frac{1}{2} \times 1\frac{3}{4}$	...
2	6	$\frac{5}{8}$	$4\frac{3}{4}$	4	$\frac{5}{8} \times 2$	...
$2\frac{1}{2}$	7	$\frac{11}{16}$	$5\frac{1}{2}$	4	$\frac{5}{8} \times 2\frac{1}{4}$	...
3	$7\frac{1}{2}$	$\frac{3}{4}$	6	4	$\frac{5}{8} \times 2\frac{1}{4}$	...
$3\frac{1}{2}$	$8\frac{1}{2}$	$\frac{13}{16}$	7	8	$\frac{5}{8} \times 2\frac{1}{2}$	...
4	9	$\frac{15}{16}$	$7\frac{1}{2}$	8	$\frac{5}{8} \times 2\frac{3}{4}$	...
5	10	$\frac{15}{16}$	$8\frac{1}{2}$	8	$\frac{3}{4} \times 2\frac{3}{4}$	...
6	11	1	$9\frac{1}{2}$	8	$\frac{3}{4} \times 3$	...
8	$13\frac{1}{2}$	$1\frac{1}{8}$	$11\frac{3}{4}$	8	$\frac{3}{4} \times 3\frac{1}{4}$	...
10	16	$1\frac{3}{16}$	$14\frac{1}{4}$	12	$\frac{7}{8} \times 3\frac{1}{2}$	...
12	19	$1\frac{1}{4}$	17	12	$\frac{7}{8} \times 3\frac{1}{2}$	...
14 O.D.	21	$1\frac{3}{8}$	$18\frac{3}{4}$	12	1×4	...
16 O.D.	$23\frac{1}{2}$	$1\frac{7}{16}$	$21\frac{1}{4}$	16	$1 \times 4\frac{1}{4}$	...
18 O.D.	25	$1\frac{9}{16}$	$22\frac{3}{4}$	16	$1\frac{1}{8} \times 4\frac{1}{2}$	...
20 O.D.	$27\frac{1}{2}$	$1\frac{11}{16}$	25	20	$1\frac{1}{8} \times 4\frac{3}{4}$	...
24 O.D.	32	$1\frac{7}{8}$	$29\frac{1}{2}$	20	$1\frac{1}{4} \times 5\frac{1}{4}$	...
30 O.D.	$38\frac{3}{4}$	$2\frac{1}{8}$	36	28	$1\frac{1}{4} \times 5\frac{3}{4}$	...
36 O.D.	46	$2\frac{3}{8}$	$42\frac{3}{4}$	32	$1\frac{1}{2} \times 6\frac{1}{2}$	...
42 O.D.	53	$2\frac{5}{8}$	$49\frac{1}{2}$	36	$1\frac{1}{2} \times 7\frac{1}{4}$	$9\frac{1}{2}$
48 O.D.	$59\frac{1}{2}$	$2\frac{3}{4}$	56	44	$1\frac{1}{2} \times 7\frac{1}{2}$	$9\frac{1}{2}$
54 O.D.	$66\frac{1}{4}$	3	$62\frac{3}{4}$	44	$1\frac{3}{4} \times 8\frac{1}{4}$	$10\frac{1}{2}$
60 O.D.	73	$3\frac{1}{8}$	$69\frac{1}{4}$	52	$1\frac{3}{4} \times 8\frac{1}{2}$	11
72 O.D.	$86\frac{1}{2}$	$3\frac{1}{2}$	$82\frac{1}{2}$	60	$1\frac{3}{4} \times 9\frac{1}{2}$	12
84 O.D.	$99\frac{3}{4}$	$3\frac{7}{8}$	$95\frac{1}{2}$	64	$2 \times 10\frac{1}{2}$	13
96 O.D.	$113\frac{1}{4}$	$4\frac{1}{4}$	$108\frac{1}{2}$	68	$2\frac{1}{4} \times 11\frac{1}{2}$	$14\frac{1}{2}$

For bolts smaller than $1\frac{3}{4}$ inches, bolt holes are drilled $\frac{1}{8}$ inch larger than bolt diameter. For bolts $1\frac{3}{4}$ inches and larger, bolt holes are drilled $\frac{1}{4}$ inch larger than bolt diameter. For bolts $1\frac{3}{4}$ inches and larger bolt studs with a nut on each end are recommended. All Flanges have plain faces.

Table A-27. - Templates for Drilling

250-Pound (Extra Heavy) Cast Iron Flanges and Fittings
A.S.A. B 16b - 1928

Size Inches	Flange Diameter Inches	Flange Thickness Inches	Bolt Circle Inches	Number of Bolts	Diameter and length of Bolts Inches	Length of Bolt Studs with 2 Nuts Inches
1	$4\frac{7}{8}$	$\frac{11}{16}$	$3\frac{1}{2}$	4	$\frac{5}{8}$ x $2\frac{1}{4}$	...
$1\frac{1}{4}$	$5\frac{1}{4}$	$\frac{3}{4}$	$3\frac{7}{8}$	4	$\frac{5}{8}$ x $2\frac{1}{2}$	...
$1\frac{1}{2}$	$6\frac{1}{8}$	$\frac{13}{16}$	$4\frac{1}{2}$	4	$\frac{3}{4}$ x $2\frac{1}{2}$	...
2	$6\frac{1}{2}$	$\frac{7}{8}$	5	8	$\frac{5}{8}$ x $2\frac{1}{2}$	...
$2\frac{1}{2}$	$7\frac{1}{2}$	1	$5\frac{7}{8}$	8	$\frac{3}{4}$ x 3	...
3	$8\frac{1}{4}$	$1\frac{1}{8}$	$6\frac{5}{8}$	8	$\frac{3}{4}$ x $3\frac{1}{4}$	...
$3\frac{1}{2}$	9	$1\frac{3}{16}$	$7\frac{1}{4}$	8	$\frac{3}{4}$ x $3\frac{1}{4}$	...
4	10	$1\frac{1}{4}$	$7\frac{7}{8}$	8	$\frac{3}{4}$ x $3\frac{1}{2}$	...
5	11	$1\frac{3}{8}$	$9\frac{1}{4}$	8	$\frac{3}{4}$ x $3\frac{3}{4}$	...
6	$12\frac{1}{2}$	$1\frac{7}{16}$	$10\frac{5}{8}$	12	$\frac{3}{4}$ x $3\frac{3}{4}$	...
8	15	$1\frac{5}{8}$	13	12	$\frac{7}{8}$ x $4\frac{1}{4}$	...
10	$17\frac{1}{2}$	$1\frac{7}{8}$	$15\frac{1}{4}$	16	1 x 5	...
12	$20\frac{1}{2}$	2	$17\frac{3}{4}$	16	$1\frac{1}{8}$ x $5\frac{1}{2}$	...
14 O.D.	23	$2\frac{1}{8}$	$20\frac{1}{4}$	20	$1\frac{1}{8}$ x $5\frac{3}{4}$	...
16 O.D.	$25\frac{1}{2}$	$2\frac{1}{4}$	$22\frac{1}{2}$	20	$1\frac{1}{4}$ x 6	...
18 O.D.	28	$2\frac{3}{8}$	$24\frac{3}{4}$	24	$1\frac{1}{4}$ x $6\frac{1}{4}$	...
20 O.D.	$30\frac{1}{2}$	$2\frac{1}{2}$	27	24	$1\frac{1}{4}$ x $6\frac{1}{2}$	...
24 O.D.	36	$2\frac{3}{4}$	32	24	$1\frac{1}{2}$ x $7\frac{1}{2}$	$9\frac{1}{2}$
30 O.D.	43	3	$39\frac{1}{4}$	28	$1\frac{3}{4}$ x $8\frac{1}{4}$	$10\frac{1}{2}$
36 O.D.	50	$3\frac{3}{8}$	46	32	2 x $9\frac{1}{4}$	$11\frac{1}{2}$
42 O.D.	57	$3\frac{11}{16}$	$52\frac{3}{4}$	36	2 x $9\frac{3}{4}$	12
48 O.D.	65	4	$60\frac{3}{4}$	40	2 x $10\frac{1}{2}$	13

All 250-pound cast iron flanges have $\frac{1}{16}$-inch raised faces. The raised face is included in the dimension for thickness of flange. For bolts smaller than $1\frac{3}{4}$ inches, bolt holes are drilled $\frac{1}{8}$ inch larger than bolt diameter. For bolts $1\frac{3}{4}$ inches and larger, bolt holes are drilled $\frac{1}{4}$ inch larger than bolt diameter. For bolts $1\frac{3}{4}$ inches and larger, bolt studs with a nut on each end are recommended.

Table A-28. - Templates for Drilling

800-Pound Cast Iron Hydraulic Flanges and Fittings
A.S.A. B 16b-1 - 1931

Size Inches	Flange Diameter Inches	Flange Thickness Inches	Bolt Circle Inches	Number of Bolts	Diameter and Length of Bolts Inches	Diameter of Raised Face Inches
2	$6\frac{1}{2}$	$1\frac{1}{4}$	5	8	$\frac{5}{8}$ x 4	$3\frac{5}{8}$
$2\frac{1}{2}$	$7\frac{1}{2}$	$1\frac{3}{8}$	$5\frac{7}{8}$	8	$\frac{3}{4}$ x $4\frac{1}{4}$	$4\frac{1}{8}$
3	$8\frac{1}{4}$	$1\frac{1}{2}$	$6\frac{5}{8}$	8	$\frac{3}{4}$ x $4\frac{1}{2}$	5
$3\frac{1}{2}$	9	$1\frac{5}{8}$	$7\frac{1}{4}$	8	$\frac{7}{8}$ x $4\frac{3}{4}$	$5\frac{1}{2}$
4	$10\frac{3}{4}$	$1\frac{7}{8}$	$8\frac{1}{2}$	8	$\frac{7}{8}$ x $5\frac{1}{4}$	$6\frac{3}{16}$
5	13	$2\frac{1}{8}$	$10\frac{1}{2}$	8	1 x 6	$7\frac{5}{16}$
6	14	$2\frac{1}{4}$	$11\frac{1}{2}$	12	1 x $6\frac{1}{4}$	$8\frac{1}{2}$
8	$16\frac{1}{2}$	$2\frac{1}{2}$	$13\frac{3}{4}$	12	$1\frac{1}{8}$ x 7	$10\frac{5}{8}$
10	20	$2\frac{7}{8}$	17	16	$1\frac{1}{4}$ x $7\frac{3}{4}$	$12\frac{3}{4}$
12	22	3	$19\frac{1}{4}$	20	$1\frac{1}{4}$ x 8	15

The 800-pound flanged valves and fittings have $\frac{1}{4}$ inch raised faces. The raised face is not included in the dimension for thickness of flange. Companion flanges are furnished with $\frac{1}{4}$ inch raised face (male) or $\frac{3}{16}$ inch (female) face as ordered.

Bolt holes are drilled $\frac{1}{8}$ inch larger than nominal diameter of bolts.

For sizes larger than 12 inches, use 600-pound steel material.

Table A-29. - Templates for Drilling

150-Pound American Standard Steel Flanges
A.S.A. B 16e - 1939

Pipe Size Inches	Flange Diameter Inches	Flange Thickness Inches	Diameter of Raised Face Inches	Bolt Circle Inches	Number of Bolts	Diameter and Length of Bolts Inches
1	$4\frac{1}{4}$	$\frac{7}{16}$	2	$3\frac{1}{8}$	4	$\frac{1}{2}$ x $1\frac{3}{4}$
$1\frac{1}{4}$	$4\frac{5}{8}$	$\frac{1}{2}$	$2\frac{1}{2}$	$3\frac{1}{2}$	4	$\frac{1}{2}$ x $1\frac{3}{4}$
$1\frac{1}{2}$	5	$\frac{9}{16}$	$2\frac{7}{8}$	$3\frac{7}{8}$	4	$\frac{1}{2}$ x 2
2	6	$\frac{5}{8}$	$3\frac{5}{8}$	$4\frac{3}{4}$	4	$\frac{5}{8}$ x $2\frac{1}{4}$
$2\frac{1}{2}$	7	$\frac{11}{16}$	$4\frac{1}{8}$	$5\frac{1}{2}$	4	$\frac{5}{8}$ x $2\frac{1}{4}$
3	$7\frac{1}{2}$	$\frac{3}{4}$	5	6	4	$\frac{5}{8}$ x $2\frac{1}{2}$
$3\frac{1}{2}$	$8\frac{1}{2}$	$\frac{13}{16}$	$5\frac{1}{2}$	7	8	$\frac{5}{8}$ x $2\frac{1}{2}$
4	9	$\frac{15}{16}$	$6\frac{3}{16}$	$7\frac{1}{2}$	8	$\frac{5}{8}$ x $2\frac{3}{4}$
5	10	$\frac{15}{16}$	$7\frac{5}{16}$	$8\frac{1}{2}$	8	$\frac{3}{4}$ x $2\frac{3}{4}$
6	11	1	$8\frac{1}{2}$	$9\frac{1}{2}$	8	$\frac{3}{4}$ x 3
8	$13\frac{1}{2}$	$1\frac{1}{8}$	$10\frac{5}{8}$	$11\frac{3}{4}$	8	$\frac{3}{4}$ x $3\frac{1}{4}$
10	16	$1\frac{3}{16}$	$12\frac{3}{4}$	$14\frac{1}{4}$	12	$\frac{7}{8}$ x $3\frac{1}{2}$
12	19	$1\frac{1}{4}$	15	17	12	$\frac{7}{8}$ x $3\frac{3}{4}$
14	21	$1\frac{3}{8}$	$16\frac{1}{4}$	$18\frac{3}{4}$	12	1 x 4
16	$23\frac{1}{2}$	$1\frac{7}{16}$	$18\frac{1}{2}$	$21\frac{1}{4}$	16	1 x $4\frac{1}{4}$
18	25	$1\frac{9}{16}$	21	$22\frac{3}{4}$	16	$1\frac{1}{8}$ x $4\frac{1}{2}$
20	$27\frac{1}{2}$	$1\frac{11}{16}$	23	25	20	$1\frac{1}{8}$ x $4\frac{3}{4}$
24	32	$1\frac{7}{8}$	$27\frac{1}{4}$	$29\frac{1}{2}$	20	$1\frac{1}{4}$ x $5\frac{1}{4}$

The thicknesses of flanges include the $\frac{1}{16}$ inch raised face.
Bolt holes are drilled $\frac{1}{8}$ inch larger than the diameter of the bolts.

Table A-30. - Templates for Drilling

300-Pound American Standard Steel Flanges
A.S.A. B 16e - 1939

Pipe Size Inches	Flange Diameter Inches	Flange Thickness Inches	Diameter of Raised Face Inches	Bolt Circle Inches	Number of Bolts	Diameter and Length of Bolts Inches
$\frac{1}{2}$	$3\frac{3}{4}$	$\frac{5}{8}$	$1\frac{3}{8}$	$2\frac{5}{8}$	4	$\frac{1}{2}$ x 2
$\frac{3}{4}$	$4\frac{5}{8}$	$\frac{11}{16}$	$1\frac{11}{16}$	$3\frac{1}{4}$	4	$\frac{5}{8}$ x $2\frac{1}{4}$
1	$4\frac{7}{8}$	$\frac{3}{4}$	2	$3\frac{1}{2}$	4	$\frac{5}{8}$ x $2\frac{1}{4}$
$1\frac{1}{4}$	$5\frac{1}{4}$	$\frac{7}{8}$	$2\frac{1}{2}$	$3\frac{7}{8}$	4	$\frac{5}{8}$ x $2\frac{1}{2}$
$1\frac{1}{2}$	$6\frac{1}{8}$	$\frac{15}{16}$	$2\frac{7}{8}$	$4\frac{1}{2}$	4	$\frac{3}{4}$ x $2\frac{3}{4}$
2	$6\frac{1}{2}$	$\frac{7}{8}$	$3\frac{5}{8}$	5	8	$\frac{5}{8}$ x $2\frac{3}{4}$
$2\frac{1}{2}$	$7\frac{1}{2}$	1	$4\frac{1}{8}$	$5\frac{7}{8}$	8	$\frac{3}{4}$ x 3
3	$8\frac{1}{4}$	$1\frac{1}{8}$	5	$6\frac{5}{8}$	8	$\frac{3}{4}$ x $3\frac{1}{4}$
$3\frac{1}{2}$	9	$1\frac{3}{16}$	$5\frac{1}{2}$	$7\frac{1}{4}$	8	$\frac{3}{4}$ x $3\frac{1}{4}$
4	10	$1\frac{1}{4}$	$6\frac{3}{16}$	$7\frac{7}{8}$	8	$\frac{3}{4}$ x $3\frac{1}{2}$
5	11	$1\frac{3}{8}$	$7\frac{5}{16}$	$9\frac{1}{4}$	8	$\frac{3}{4}$ x $3\frac{3}{4}$
6	$12\frac{1}{2}$	$1\frac{7}{16}$	$8\frac{1}{2}$	$10\frac{5}{8}$	12	$\frac{3}{4}$ x 4
8	15	$1\frac{5}{8}$	$10\frac{5}{8}$	13	12	$\frac{7}{8}$ x $4\frac{1}{2}$
10	$17\frac{1}{2}$	$1\frac{7}{8}$	$12\frac{3}{4}$	$15\frac{1}{4}$	16	1 x 5
12	$20\frac{1}{2}$	2	15	$17\frac{3}{4}$	16	$1\frac{1}{8}$ x $5\frac{1}{2}$
14	23	$2\frac{1}{8}$	$16\frac{1}{4}$	$20\frac{1}{4}$	20	$1\frac{1}{8}$ x $5\frac{3}{4}$
16	$25\frac{1}{2}$	$2\frac{1}{4}$	$18\frac{1}{2}$	$22\frac{1}{2}$	20	$1\frac{1}{4}$ x 6
18	28	$2\frac{3}{8}$	21	$24\frac{3}{4}$	24	$1\frac{1}{4}$ x $6\frac{1}{4}$
20	$30\frac{1}{2}$	$2\frac{1}{2}$	23	27	24	$1\frac{1}{4}$ x $6\frac{1}{2}$
24	36	$2\frac{3}{4}$	$27\frac{1}{4}$	32	24	$1\frac{1}{2}$ x $7\frac{1}{4}$

The thicknesses of flanges include the $\frac{1}{16}$ inch raised face.
Bolt holes are drilled $\frac{1}{8}$ inch larger than the diameter of the bolts.

Table A-31. - Templates for Drilling

400-Pound American Standard Steel Flanges
A.S.A. B 16e - 1939

For sizes $3\frac{1}{2}$ inches and smaller,
use 600-Pound American Standard

Pipe Size Inches	Flange Diameter Inches	Flange Thickness Inches	Diameter of Raised Face Inches	Bolt Circle Inches	Number of Bolts	Diameter and Length of Bolts Inches
4	10	$1\frac{3}{8}$	$6\frac{3}{16}$	$7\frac{7}{8}$	8	$\frac{7}{8}$ x $5\frac{1}{4}$
5	11	$1\frac{1}{2}$	$7\frac{5}{16}$	$9\frac{1}{4}$	8	$\frac{7}{8}$ x $5\frac{1}{2}$
6	$12\frac{1}{2}$	$1\frac{5}{8}$	$8\frac{1}{2}$	$10\frac{5}{8}$	12	$\frac{7}{8}$ x $5\frac{3}{4}$
8	15	$1\frac{7}{8}$	$10\frac{5}{8}$	13	12	1 x $6\frac{1}{2}$
10	$17\frac{1}{2}$	$2\frac{1}{8}$	$12\frac{3}{4}$	$15\frac{1}{4}$	16	$1\frac{1}{8}$ x $7\frac{1}{4}$
12	$20\frac{1}{2}$	$2\frac{1}{4}$	15	$17\frac{3}{4}$	16	$1\frac{1}{4}$ x $7\frac{3}{4}$
14	23	$2\frac{3}{8}$	$16\frac{1}{4}$	$20\frac{1}{4}$	20	$1\frac{1}{4}$ x 8
16	$25\frac{1}{2}$	$2\frac{1}{2}$	$18\frac{1}{2}$	$22\frac{1}{2}$	20	$1\frac{3}{8}$ x $8\frac{1}{2}$
18	28	$2\frac{5}{8}$	21	$24\frac{3}{4}$	24	$1\frac{3}{8}$ x $8\frac{3}{4}$
20	$30\frac{1}{2}$	$2\frac{3}{4}$	23	27	24	$1\frac{1}{2}$ x $9\frac{1}{2}$
24	36	3	$27\frac{1}{4}$	32	24	$1\frac{3}{4}$ x $10\frac{1}{4}$

The thicknesses of flanges do not include the $\frac{1}{4}$ inch raised or grooved face.

Bolt holes are drilled $\frac{1}{8}$ inch larger than the diameter of the bolts.

Table A-32. - Templates for Drilling

600-Pound American Standard Steel Flanges
A.S.A. B 16e - 1939

Pipe Sizes Inches	Flange Diameter Inches	Flange Thickness Inches	Diameter of Raised Face Inches	Bolt Circle Inches	Number of Bolts	Diameter and Length of Bolts Inches
$\frac{1}{2}$	$3\frac{3}{4}$	$\frac{9}{16}$	$1\frac{3}{8}$	$2\frac{5}{8}$	4	$\frac{1}{2}$ x 3
$\frac{3}{4}$	$4\frac{5}{8}$	$\frac{5}{8}$	$1\frac{11}{16}$	$3\frac{1}{4}$	4	$\frac{5}{8}$ x $3\frac{1}{4}$
1	$4\frac{7}{8}$	$\frac{11}{16}$	2	$3\frac{1}{2}$	4	$\frac{5}{8}$ x $3\frac{1}{2}$
$1\frac{1}{4}$	$5\frac{1}{4}$	$\frac{13}{16}$	$2\frac{1}{2}$	$3\frac{7}{8}$	4	$\frac{5}{8}$ x $3\frac{3}{4}$
$1\frac{1}{2}$	$6\frac{1}{8}$	$\frac{7}{8}$	$2\frac{7}{8}$	$4\frac{1}{2}$	4	$\frac{3}{4}$ x 4
2	$6\frac{1}{2}$	1	$3\frac{5}{8}$	5	8	$\frac{5}{8}$ x 4
$2\frac{1}{2}$	$7\frac{1}{2}$	$1\frac{1}{8}$	$4\frac{1}{8}$	$5\frac{7}{8}$	8	$\frac{3}{4}$ x $4\frac{1}{2}$
3	$8\frac{1}{4}$	$1\frac{1}{4}$	5	$6\frac{5}{8}$	8	$\frac{3}{4}$ x $4\frac{3}{4}$
$3\frac{1}{2}$	9	$1\frac{3}{8}$	$5\frac{1}{2}$	$7\frac{1}{4}$	8	$\frac{7}{8}$ x $5\frac{1}{4}$
4	$10\frac{3}{4}$	$1\frac{1}{2}$	$6\frac{3}{16}$	$8\frac{1}{2}$	8	$\frac{7}{8}$ x $5\frac{1}{2}$
5	13	$1\frac{3}{4}$	$7\frac{5}{16}$	$10\frac{1}{2}$	8	1 x $6\frac{1}{2}$
6	14	$1\frac{7}{8}$	$8\frac{1}{2}$	$11\frac{1}{2}$	12	1 x $6\frac{3}{4}$
8	$16\frac{1}{2}$	$2\frac{3}{16}$	$10\frac{5}{8}$	$13\frac{3}{4}$	12	$1\frac{1}{8}$ x $7\frac{1}{2}$
10	20	$2\frac{1}{2}$	$12\frac{3}{4}$	17	16	$1\frac{1}{4}$ x $8\frac{1}{4}$
12	22	$2\frac{5}{8}$	15	$19\frac{1}{4}$	20	$1\frac{1}{4}$ x $8\frac{1}{2}$
14	$23\frac{3}{4}$	$2\frac{3}{4}$	$16\frac{1}{4}$	$20\frac{3}{4}$	20	$1\frac{3}{8}$ x 9
16	27	3	$18\frac{1}{2}$	$23\frac{3}{4}$	20	$1\frac{1}{2}$ x $9\frac{3}{4}$
18	$29\frac{1}{4}$	$3\frac{1}{4}$	21	$25\frac{3}{4}$	20	$1\frac{5}{8}$ x $10\frac{1}{2}$
20	32	$3\frac{1}{2}$	23	$28\frac{1}{2}$	24	$1\frac{5}{8}$ x 11
24	37	4	$27\frac{1}{4}$	33	24	$1\frac{7}{8}$ x $12\frac{1}{2}$

The thicknesses of flanges do not include the $\frac{1}{4}$ inch raised or grooved face.

Bolt holes are drilled $\frac{1}{8}$ inch larger than the diameter of the bolts.

Table A-33. – Templates for Drilling

900-Pound American Standard Steel Flanges
A.S.A. B 16e - 1939

Pipe Sizes Inches	Flange Diameter Inches	Flange Thickness Inches	Diameter of Raised Face Inches	Bolt Circle Inches	Number of Bolts	Diameter and Length of Bolts Inches
For sizes $2\frac{1}{2}$ inches and smaller, use 1,500-Pound American Standard						
3	$9\frac{1}{2}$	$1\frac{1}{2}$	5	$7\frac{1}{2}$	8	$\frac{7}{8}$ x $5\frac{1}{2}$
4	$11\frac{1}{2}$	$1\frac{3}{4}$	$6\frac{3}{16}$	$9\frac{1}{4}$	8	$1\frac{1}{8}$ x $6\frac{1}{2}$
5	$13\frac{3}{4}$	2	$7\frac{5}{16}$	11	8	$1\frac{1}{4}$ x $7\frac{1}{2}$
6	15	$2\frac{3}{16}$	$8\frac{1}{2}$	$12\frac{1}{2}$	12	$1\frac{1}{8}$ x $7\frac{1}{2}$
8	$18\frac{1}{2}$	$2\frac{1}{2}$	$10\frac{5}{8}$	$15\frac{1}{2}$	12	$1\frac{3}{8}$ x $8\frac{1}{2}$
10	$21\frac{1}{2}$	$2\frac{3}{4}$	$12\frac{3}{4}$	$18\frac{1}{2}$	16	$1\frac{3}{8}$ x 9
12	24	$3\frac{1}{8}$	15	21	20	$1\frac{3}{8}$ x $9\frac{3}{4}$
14	$25\frac{1}{4}$	$3\frac{3}{8}$	$16\frac{1}{4}$	22	20	$1\frac{1}{2}$ x $10\frac{1}{2}$
16	$27\frac{3}{4}$	$3\frac{1}{2}$	$18\frac{1}{2}$	$24\frac{1}{4}$	20	$1\frac{5}{8}$ x 11
18	31	4	21	27	20	$1\frac{7}{8}$ x $12\frac{1}{2}$
20	$33\frac{3}{4}$	$4\frac{1}{4}$	23	$29\frac{1}{2}$	20	2 x $13\frac{1}{4}$
24	41	$5\frac{1}{2}$	$27\frac{1}{4}$	$35\frac{1}{2}$	20	$2\frac{1}{2}$ x $16\frac{3}{4}$

The thicknesses of flanges do not include the $\frac{1}{4}$ inch raised or grooved face.

Bolt holes are drilled $\frac{1}{8}$ inch larger than the diameter of the bolts.

Table A-34. - Templates for Drilling

1,500-Pound American Standard Steel Flanges
A.S.A. B 16e - 1939

Pipe Sizes Inches	Flange Diameter Inches	Flange Thickness Inches	Diameter of Raised Face Inches	Bolt Circle Inches	Number of Bolts	Diameter and Length of Bolts Inches
1	$5\frac{7}{8}$	$1\frac{1}{8}$	2	4	4	$\frac{7}{8}$ x $4\frac{3}{4}$
$1\frac{1}{4}$	$6\frac{1}{4}$	$1\frac{1}{8}$	$2\frac{1}{2}$	$4\frac{3}{8}$	4	$\frac{7}{8}$ x $4\frac{3}{4}$
$1\frac{1}{2}$	7	$1\frac{1}{4}$	$2\frac{7}{8}$	$4\frac{7}{8}$	4	1 x $5\frac{1}{4}$
2	$8\frac{1}{2}$	$1\frac{1}{2}$	$3\frac{5}{8}$	$6\frac{1}{2}$	8	$\frac{7}{8}$ x $5\frac{1}{2}$
$2\frac{1}{2}$	$9\frac{5}{8}$	$1\frac{5}{8}$	$4\frac{1}{8}$	$7\frac{1}{2}$	8	1 x 6
3	$10\frac{1}{2}$	$1\frac{7}{8}$	5	8	8	$1\frac{1}{8}$ x $6\frac{3}{4}$
4	$12\frac{1}{4}$	$2\frac{1}{8}$	$6\frac{3}{16}$	$9\frac{1}{2}$	8	$1\frac{1}{4}$ x $7\frac{1}{2}$
5	$14\frac{3}{4}$	$2\frac{7}{8}$	$7\frac{5}{16}$	$11\frac{1}{2}$	8	$1\frac{1}{2}$ x $9\frac{1}{2}$
6	$15\frac{1}{2}$	$3\frac{1}{4}$	$8\frac{1}{2}$	$12\frac{1}{2}$	12	$1\frac{3}{8}$ x 10
8	19	$3\frac{5}{8}$	$10\frac{5}{8}$	$15\frac{1}{2}$	12	$1\frac{5}{8}$ x $11\frac{1}{4}$
10	23	$4\frac{1}{4}$	$12\frac{3}{4}$	19	12	$1\frac{7}{8}$ x 13
12	$26\frac{1}{2}$	$4\frac{7}{8}$	15	$22\frac{1}{2}$	16	2 x $14\frac{1}{2}$

The thicknesses of flanges do not include the $\frac{1}{4}$ inch raised or grooved face.

Bolt holes are drilled $\frac{1}{8}$ inch larger than the diameter of the bolts.

Table A-35.--Maximum Pressure-Temperature Ratings for Flanges at Temperatures from 100° to 1000°F. for Power and Oil Piping.

Primary Service Pressure Ratings	150	300	400	600	900	1,500
Maximum Hydrostatic Shell Test Pressures	350	750	1,000	1,500	2,000	3,500
Service Temperatures Degrees Fahr.	Maximum Non-Shock Service Pressures, Lb. per Sq. In.					
	Power Piping and Oil Piping					
100	230	500	670	1,000	1,500	2,500
150	220	480	640	960	1,440	2,400
200	210	465	620	930	1,395	2,325
250	200	450	600	900	1,350	2,250
300	190	435	580	870	1,305	2,175
350	180	420	560	840	1,260	2,100
400	170	405	540	810	1,215	2,025
450	160	390	520	780	1,170	1,950
500	150	375	500	750	1,125	1,875
550	140	360	480	720	1,080	1,800
600	130	345	460	690	1,035	1,725
650	120	330	440	660	990	1,650
700	110	315	420	630	950	1,575
750	100	300	400	600	900	1,500
	Power Piping					
800	85	250	335	500	750	1,250
850	70	200	270	400	600	1,000
	Oil Piping					
800	92	275	370	550	830	1,380
850	82	245	330	490	740	1,230
900	70	210	280	420	630	1,050
950	55	165	220	330	495	825
1,000	40	120	160	240	360	600

Hydrostatic tests shall be made with water at a temperature not to exceed 125°F.

Table A-36. - Type Numbers of Stainless Steels
(American Iron and Steel Institute)

Type No.	Carbon	Chrome	Nickel	Other Elements
301X	.10-.20	16.00-17.50	7.00- 8.50-	
302	over .08-.20	17.50-19.00	8.00- 9.00	
302B	over .08-.20	17.50-19.00	8.00- 9.00	Si 2.00-3.00
303	.20 max.	17.50-19.00	8.00- 9.00	S or Se .07 min. or Mo .60
304	.08 max.	17.50-19.00	8.00- 9.00	
305	over .08-.20	18.00-20.00	9.00-10.00	
306	.08 max.	18.00-20.00	9.00-10.00	
307	over .08-.20	20.00-22.00	10.00-12.00	
308	.08 max.	20.00-22.00	10.00-12.00	
309	.20 max.	22.00-26.00	12.00-14.00	
310	.25 max.	24.00-26.00	19.00-21.00	
311	.25 max.	19.00-21.00	24.00-26.00	
312	.25 max.	27.00-31.00	8.00-10.00	
315	.15 max.	17.00-19.00	7.00- 9.50	Cu 1.00-1.50, Mo 1.00-1.50
316	.10 max.	16.00-19.00	14.00 max.	Mo 2.00-3.00
317	.10 max.	18.00-20.00	12.00-14.00	Mo 3.00-4.00
321	.10 max.	17.00-20.00	7.00-10.00	Ti min. 4 x C
325	.25 max.	7.00-10.00	19.00-23.00	Cu 1.00-1.50
327	.25 max.	25.00-30.00	3.00- 5.00	
329	.10 max.	25.00-30.00	3.00- 5.00	Mo 1.00-1.50
330	.25 max.	14.00-16.00	33.00-36.00	
343	over.25	12.00-16.00	12.00-16.00	W 3.00
347	.10 max.	17.00-20.00	8.00-12.00	Cb 10 x C
403	.12 max.	11.50-13.00		Turbine Quality
405	.08 max.	11.50-13.50		Al .10-.20
406	.12 max.	12.00-14.00		Al 4.00-4.50
410	.12 max.	10.00-13.50		
414	.12 max.	10.00-13.50	2.00 max.	
416	.12 max.	12.00-14.00		S or Se .07 min or Mo .60 max.
418	.12 max.	12.00-14.00		W 2.50-3.50
420	over.12	12.00-14.00		
420F	" .12	12.00-14.00		S or Se .07min or Mo .60 max.
430	.12 max.	14.00-18.00		
430F	.12 max.	14.00-18.00		S or Se .07 min or Mo .60 max.
431	.15 max.	14.00-18.00	2.00 max.	
434A	.12 max.	14.00-18.00		Si 1.00, Cu 1.00
438	.12 max.	16.00-18.00		W 2.50-3.50
439	.50-.65	8.00		W 8.00
440	over .12	14.00-18.00		
441	" .15	14.00-18.00	2.00 max.	
442	.35 max.	18.00-23.00		
446	.35 max.	23.00-30.00		
501	" .10	4.00-6.00		
502	.10 max.	4.00-6.00		

Table A-37. - Specifications for Carbon Steels and Alloy Steels

Specification ASTM	 ASME	Gra	Composition (partial)%
Plate Steels			
Carbon Steels			
A 70	S-1		C 0.35 max, Mn 0.80
A 89	S-2	A	C 0.15 max., Mn 0.35-0.60
A 89	S-2	B	C 0.20 max., Mn 0.35-0.60
A 201	S-42	A	C 0.20-0.31, Mn 0.80, Si 0.15-0.30
A 201	S-42	B	C 0.24-0.35, Mn 0.80, Si 0.15-0.30
Low-Alloy Steels			
A 202	S-28	A	C 0.17, Cr 0.30-0.60, Mn 1.05-1.40, Si 0.6-0.9
A 202	S-28	B	C 0.25, Cr 0.3-0.6, Mn 1.05-1.4, Si 0.6-0.9
A 203	S-43	A	C 0.17, Mn 0.8, Ni 2.0-2.75, Si 0.15-0.30
A 203	S-43	B	C 0.20 ditto
A 203	S-43	C	C 0.25, "
A 204	S-44	A	C 0.18-0.25, Mn 0.5-0.9, Si 0.15-0.30, Mo 0.4-0.6
A 204	S-44	B	C 0.20-0.27, ditto
A 204	S-44	C	C 0.23-0.28, ditto
Forgings			
Carbon Steels			
	S-4	1	C 0.35, Mn 0.4-0.9, Si 0.15-0.3
	S-4	2	C 0.35, Mn 0.4-0.9, Si 0.15-0.3
	S-4	3	C 0.50, Mn 0.5-0.9 Si 0.3 max
A 105	S-8	1 2	Mn 0.4-0.8
A 18	S-10	A	Mn 0.3-0.6
A 18	S-10	B C	Mn 0.4-0.8
A 181	S-50	1	C 0.35, Mn 0.4-0.8
Alloy Steels			
A182	S-35	F1	C 0.35, Mn 0.3-0.8, Si 0.2-0.5, Mo 0.4-0.6
A182	S-35	F3	C 0.15-0.25, Mn 0.4-0.6, Si 0.45-0.75, Cr 1.5-2.0, Mo 0.6-0.8
A182	S-35	F4	C 0.35-0.45, Mn 0.5-0.8, Ni 1.5-2.0, Cr 0.5-0.8, Mo 0.3-0.4
A182	S-35	F5	C 0.15-0.25, Mn 0.3-0.5, Cr 4.0-6.0, Mo 0.45- 0.65, Ti 1.0
A182	S-35	F6	C 0.12, M 0.5, Si 0.5, Ni 0.5, Cr 11.5-13.0
A182	S-35	F7	C 0.25-0.35, Mn 0.4-0.6, Si 0.15-0.45, Cr 0.8-1.1, Mo 0.15-0.25
A182	S-35	F8	C 0.07, Mn 0.5, Si 0.2-0.4, Ni 7.0-10.0, Cr 17.0-20.0
A182	S-35	F11	C 0.25-0.40, Mn 1.2-1.5, Si 0.4-0.6, Cr 0.6- 0.9, Mo 0.25-0.4

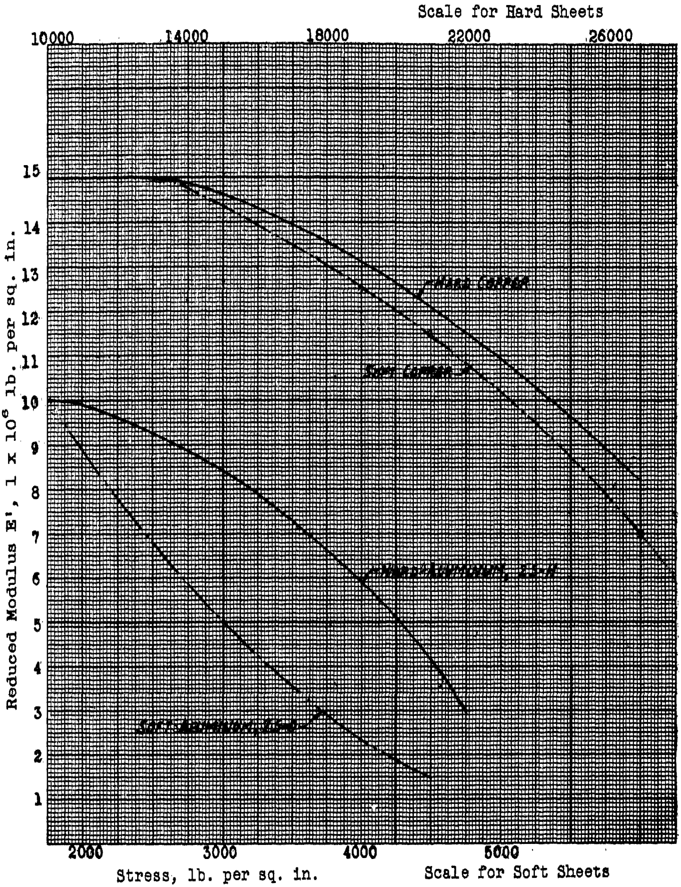


Fig. 183. Reduced Modulus E' for Aluminum and Copper Sheets.

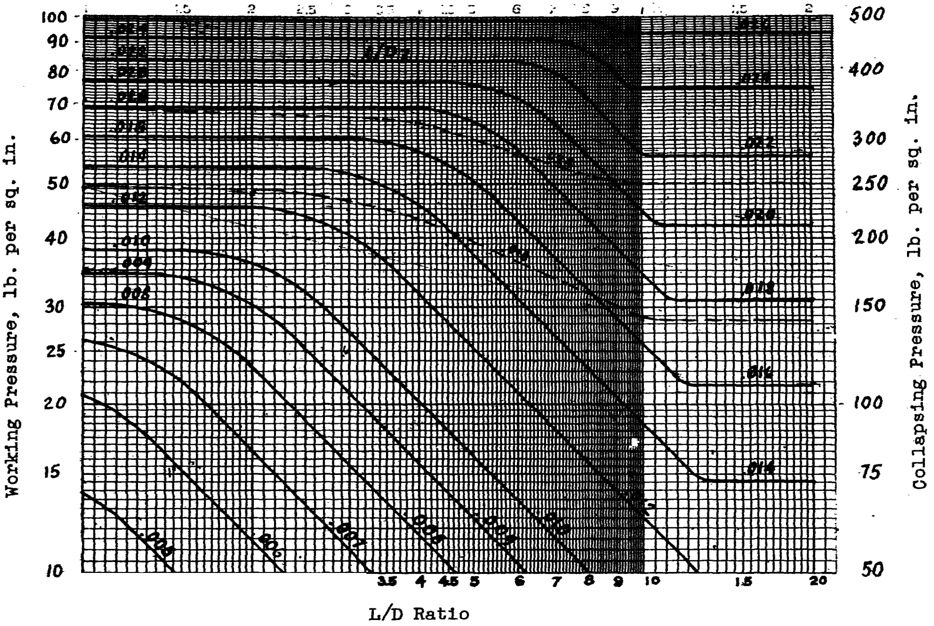


Fig. 184. Chart for Determining Shell Thickness of Cylindrical Vessels Subjected to External Pressure -- Copper.

Note: These curves have been constructed with a constant modulus E (see p. 145) and are applicable to hard copper only. Alternate curves (dash lines), which are based on the reduced modulus E' of soft copper, are shown for t/D ratios .018 and .024.

SUPPLEMENT

Design of Hoops. - Hoops around vertical tanks are placed so that each carries the same load, with the exception of the hoop at the top and bottom which, when placed at the very top and bottom of the tank, receive only half the stress taken by the intermediate hoops. Under these assumptions, and if the hoops are designed to stand only the liquid pressure within the tank, the number of required hoops N is

$$N = \frac{h^2 w r}{2 f} + 1 \qquad (1)$$

where h height of the tank, ft
r radius of the tank, ft
f allowable tension in the hoop, lb.
(= area of rod x working stress)
w = unit weight of the liquid, lb. per cu. ft.

The hoop tension f has to be adjusted so that N becomes a whole number.

The distance y from the top of the tank to any particular hoop is

$$y = h\sqrt{n/N} \qquad (2)$$

where N = total number of hoops below the top one
n = serial number of any particular hoop being located, counting the top as 0, the next one below as 1, etc.

The load per hoop (except top and bottom hoop), in lb., is

$$W = d \cdot w \cdot h^2/2N \qquad (3)$$

where d = diameter of tank, ft.

If gas or other pressure is applied in addition to the pressure of the liquid, the calculation is carried out for the height h_1 of a hypothetical open tank, in which the liquid pressure at a height h (= height of real tank) from the bottom is equal to the known pressure p at the top of the real tank. Since 1 lb. per sq. in. equals 2.309 ft. of water column, the pressure p, lb. per sq. in., is easily converted into the equivalent h_p of liquid head, in feet, $h_p = p \times 2.309/ g$, where g is the specific gravity of the liquid. The liquid head, in feet, plus the height h of the tank equals the height h_1 of the hypothetical tank, $h_1 = h + h_p$. The number of hoops and the spacing is then calculated as above. The serial number of the hoop at the top of the real tank is

$$n = (h_1 - h)^2 \cdot N/h_1^2 \qquad (4)$$

The top hoop should always fall exactly at the top of the tank. If this does not occur, a second somewhat shorter hypothetical tank with the height h_2 is calculated as follows:

$$h_2 = \frac{h}{1 - \sqrt{n_1/N}} \qquad (5)$$

where h = height of real tank
N = total number of hoops required for first hypothetical tank (h_1) below the top one
n_1 = serial number of first hoop above the top of the real tank (h)

The distance from the top of the shorter hypothetical tank to the top of the real tank is then according eq. (2): $y = h_2\sqrt{n_1/N}$. The remaining hoops are located at the following distance from the top of the real tank: $y = h_2\sqrt{n/N} - (h_2 - h)$.

Examples of Calculations.

1) 1000 gal. Still. Given: 60" I.D. jacketed still with hemispherical bottom and elliptical head as per Fig. 186.
Material: shell, carbon steel, A.S.T.M. specification A70, flange A105, Class II, bolts, alloy steel, A.S.T.M. spec. A96, Grade B, factor of safety 5, efficiency of joints 80%. Operating conditions: full vacuum in still, 150 psi steam pressure in jacket, working temperature 365°F. Still to be lined inside with 1/16" stainless steel.

Required: Design in accordance with A.S.M.E. Code.

Solution: Upper section of shell: $L/D = 32/60.75 = 0.527$, for 15 psi $t/D = .003$ (from Fig. 2), $t = .003 \times 60.75 = 0.182$, minimum allowed $t = 3/8"$. Lower section of shell: $L/D = 24/62.0 = 0.387$, for 165 psi $t/D = 0.0159$, $t = 0.0159 \times 62.0 = .984$, approx. 1.0". Bottom: $t = 5 \times 165 \times 30/2 \times 55000 \times 0.6 = 0.375"$. Cover: $a/b = 2.0$, $V = 1.0$ (Fig. 12), $t = 15 \times 60.0 \times 1.0/2 \times 11000 = 0.041$, 1/8 additional thickness to allow for manhole, minimum $t = 3/8"$. No credit to be taken for strength of stainless steel liner.

Shell of jacket: $t = 150 \times 33/11000 \times 0.8 = .562 = 9/16"$ plus 1/16" corrosion allowance. Bottom of jacket: t = 8.33 x 150 x 60/2 x 55000 ·0.682 = 11/16" plus 1/16" corr. allow.

Supports: Weight of full still incl. agitator and drive 20000 lbs., Wt. per lug 5000 lbs., a/L = 7/12 = .584, K = 1.86 (Fig. 178), R_{max} = 5000 x 1.86/12 = 774 lb. per lin. in., allow. load of 3/8" fillet weld 1420 lb. per in. (Table 45).

Flange: Select 61" I.D. x 63" O.D. asbestos gasket, 60-1" bolts on $64\frac{1}{2}$" B.C., 67" O.D. flange, gasket material factors m = 2.5, y = 4500, b = 0.5 (Table 15a). H_{Gy} = 0.5 π 62 x 4500 = 438000, H_{Gp} = 2 x 0.5 π 62 x 2.5. x 30 = 14600, H = 62^2 x 0.785 x 30 = 90800, W_{min} = H_{Gy} = 438000, W_{act} = 0.551 x 60 x 15000 = 496000, W = $\frac{1}{2}$ (438000 + 496000) = 467000, lever arm (approx.) = $\frac{1}{2}$ (64.5-62) = $1\frac{1}{4}$", M_o = 467000 x 1.25 = 582000, M = 582000/60.75 = 9600, K = 67/60.75 = 1.105, Y = 20 (Fig. 36), t = $\sqrt{9600 \times 20/14000}$ = 3.71" = 3 3/4".

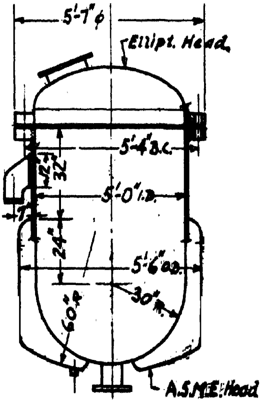


Fig. 186. Still

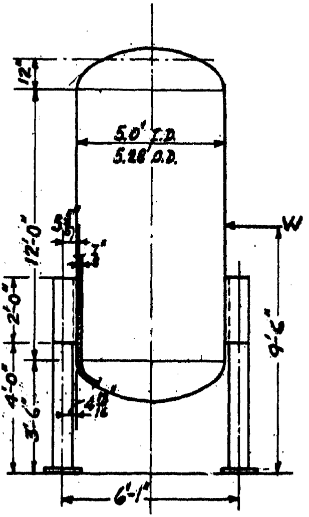


Fig. 187. Knock-out Drum

2) 5 ft. x 12 ft. Knock-out Drum. Given: 5' I.D. x 12' high knock-out drum with elliptical heads as per Fig. 187.
Material: carbon steel A.S.T.M. spec. A70, allow. stress 13750, radiographed and stress relieved. Design pressure 660 psi, design temperature 100°F, corrosion allowance 1/8", factor of safety 4, joint efficiency 95%.

Required: Design in accordance with A.P.I.-A.S.M.E. Code.

Solution: Shell thickness t = 660 x 60.25/ (27500 x 0.95-660) = 1.56, plus 0.125 = 1.685, approx. 1 11/16" Max. allow. pressure (new and cold) = 27500 x 0.95 x 1.6875/61.6875 = 715 psi; hydrostatic test pressure 715 x 1.5 = 1072 psi. Weight of drum, empty 18815 lbs., filled with water 35615 lbs.
Anchor bolts: wind load = 5.28 x 14 x 25 = 1845 lb., uplift per leg = (1845 x 9.5/6.083)-(18815/4) = - 1814 lb., 1-3/4" bolt: 0.302 x 11000 = 3320 lb. allow. stress.
Support: Max. load = (36000/4) + (1845 x 9.5/6.083) = 11790 lb., a/L = 4.8125/24 = 0.201, K = 0.9 (Fig. 178), load on weld = 11790 x 0.9/24 = 442 lb. per in., allow load on ½" fillet weld = 1900 lb. per in.

Stresses due to eccentric load: Select 8 x 6½ WF27 for legs. S = (11790/7.93) + (11790 x 5.625 x 3.26/20.8) = 1490 + 10350 = 11840 psi; allow. stress $f = 18000/\ 1 + (48^2/18000 \times 1.62^2) = 17200$ psi.

3) 4 ft. x 55 ft. Absorber Tower. Given: 4'I.D. x 55' high absorber tower with 21 bell plates and manholes as per Fig. 188. Material: Carbon steel A.S.T.M. spec. A 70, allow. stress 13750 psi, stress relieved, corrosion allowance 1/8", shell thickness 7/16", working pressure 150 psi, temperature 450°F., thickness of insulation 3".

Required: Calculation of stresses due to wind load and design of skirt.

Solution: Weights: shell (without bell plates) 19800, bell plates 10500, insulation 4900, water 50600 lbs. Stresses in shell: due to wind load $S_w = 0.0533 \times 25 \times 56.0^2 \times 4.57/4.07^2 \times 0.4375 = 2770$ psi, due to dead load (insulated and empty) $S_{de} = 24700/48.44\ \pi\ 0.4375 = 370$ psi, due to dead load (full) $S_{df} = 85800/48\ 44\ \pi\ 0.4375 = 1290$ psi, due to internal pressure $S_p = 48 \times 150/4 \times 0.4375 = 4120$ psi, max.

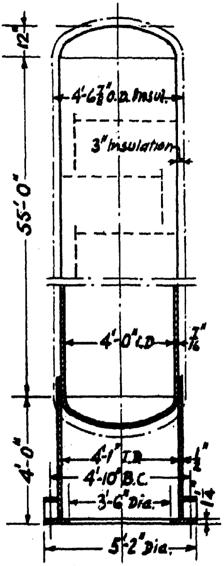


Fig. 188. Absorber Tower

compression S_c = 2770 + 1290 = 4060 psi, max. tension S_t = 2770 + 4120 - 370 = 6520 psi.

Skirt weld: $S_{max} = (0.0533 \times 25 \times 56.0^2 \times 4.57/4.07^2) + (85800/48.875\pi) = 1210 + 558 = 1768$ lb./in., efficiency of weld = 0.55 x 0.8 x 1.06 = 0.466, allow. stress of weld (½" fillet) = 0.5 x 0.707 x 13750 x 0.466 = 2260 lb./in.

Anchor bolts: assume 8 bolts on 58" B.C. spacing of bolts = $58\pi/8$ = 27.8" Load per bolt $P_1 = (0.0533 \times 25 \times 60.0^2 \times 4.57 \times 27.8/4.83^2) - (19800 \times 27\ 8/58\pi) = 26600 - 3000 = 23600$ lbs., bolt area = 23600/13750 = 1.72, say 1 3/4" bolt.
Base plate; assume 42" I.D. x 62" O.D., max. load $P' = (0.0533 \times 25 \times 60.0^2 \times 4.57/4.83^2) + (85800/58\pi) = 957 + 473 = 1430$, bearing pressure w = 1430/10 = 143 psi, projection of base plate 6". Thickness of base plate $a = 6\sqrt{3 \times 143/11000} = 1.19"$, say 1 ¼".

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